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A Review of the Heat Pump Research and Development Program

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A REVIEW OF THE HEAT PUMP RESEARCH AND DEVELOPMENT PROGRAM

A Program of the Division of Energy
National Research Council of Canada

Final Report

Prepared for:

Division of Energy
National Research Council of Canada

Prepared by:

Energy Systems Centre
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June, 1986

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ABSTRACT

This document provides an overview of the Program of Heat Pump Research and Development, conducted under the aegis of the Division of Energy, National Research Council of Canada. A brief review chapter provides a history of the NRC heat pump research program over the past ten years, and establishes a commercial and technical perspective of heat pump applications in Canada. Subsequently, summaries are provided for about twenty industrial project studies, both analytical and experimental. Finally, a similar set of summaries are provided for residential heat pump studies, both analytical and experimental.

THE HEAT PUMP RESEARCH AND DEVELOPMENT PROGRAM

The escalation of world oil prices which began in 1973 started an international search for alternative sources of energy. Application of energy conservation technology was to become a very important element in alleviation of the supply situation. The heat pump was one technology which could be applied to energy conservation and as a result, a program of heat pump research and development was begun by the National Research Council of Canada.

THE BEGINNING

The Division of Building Research (DBR) initiated, in 1976, a state-of-the-art review of residential heat pumps and their applications. At that time, residential heat pump applications were being studied by the electrical utilities (Ontario Hydro and Hydro Quebec) and the Canadian Electrical Association. The Canadian Standards Association was engaged in developing both a heat pump performance standard and an installation guideline for electrically driven, reversible air-source heat pumps. The DBR research program was planned to complement these other efforts by undertaking work not being addressed at the time.

The objectives of the DBR program were to assess the technical and economic feasibility of using heat pumps to provide part or all of the heating requirements (including space heating, domestic hot water heating and ventilation heat recovery) in residential buildings.

The DBR research activities were divided into two areas:

- Current heat pump technology (electrically driven air-source heat pumps);
- Medium to long term future developments.

Current Heat Pump Technology research activities were to determine:

- whether commercially available residential heat pumps were an economically viable alternative to conventional and non-conventional heating systems and to energy conservation techniques;
- how well commercially available residential heat pumps were satisfying the seasonal heating needs of houses, in terms of energy savings and reliability;
- the type of technical data and analytical methods required to permit a simple prediction of seasonal thermal performance of a heat pump in a residential application;
- the benefits of special design features of commercially available heat pumps on cold-weather thermal performance.

Medium to Long Term Future Development research activities were to determine:

- the feasibility of using heat sources other than outdoor air for residential heat pumps;
- the technical and economic feasibility of using heat pumps for other functions in a home (e.g. domestic water heating, ventilation heat recovery, dehumidification or a combination of functions);
- the feasibility of integrating heat pumps into other residential space heating concepts, such as district heating;
- the performance of prototype systems based on the foregoing advanced concepts.

EXPANSION OF THE PROGRAM

Following the second world oil-price shock in 1979, the need to examine heat pump applications in industry became an important element of the National Research Council of Canada activities. The Energy Research and Development Program Branch of the National Research Council of Canada (subsequently the Division of Energy) initiated projects to monitor and analyze data from industrial heat pump installations in Canada. The Division of Mechanical Engineering set about to develop a heat pump components and systems design and development program.

THE DIVISION OF ENERGY

In 1981, the overall coordination of the Heat Pump Research activities in the Divisions of Building Research and Mechanical Engineering and the new extension to the industrial sector became the responsibility of the Division of Energy.

The Division of Energy's initial task was to seek and implement as quickly as possible projects which would demonstrate the heat pump's energy conservation potential. To measure the interest and level of skills in heat pump design and application, the private sector was invited to submit proposals in a number of areas in residential, industrial and commercial applications. Review of the response resulted in a program organization with three elements: Industrial Applications; Residential Applications and System and Component Development (see Figure 1).

TERMINATION OF THE PROGRAM

Although a rigid time base for the program was not established, it was assumed that within a five year period results could be obtained and evaluated from a number of varied industrial applications and that the residential applications sector would be well understood and commercialized with new high-efficiency developments. The termination of the program in November 1984 largely frustrated this objective. Measures have now been taken to maintain the research activity, and to complete in a less rapid fashion some of the projects of future technological and economic value.

THE INTERNATIONAL ENERGY AGENCY (IEA)

The IEA is an organization set up by the Organization for Economic Co-operation and Development to promote energy conservation research. The heat pump has been identified as an important end-user component with significant potential in energy conservation. An Executive Committee has been established by the IEA with the responsibility for the coordination of international research in this field under an agreement entitled "A Program of Research and Development on Advanced Heat Pump Systems". Canada is a signatory to the implementing agreement which provides the committee with its legal basis for cooperative activity. For some years Canada continued with limited participation, specifically in the area of residential applications, until the formation of the Division of Energy in 1981. The Division of Energy provided the Canada Delegate from December 1981 and proceeded to increase Canadian participation and to identify the most advantageous interface with the working program.

THE WORKING ANNEXES

Scientific and technical business is conducted by the Executive Committee through a number of working 'annexes'. Canada has participated in a number of the annexes in the past and is at the time of writing the operating agent for Annex VIII, Advanced In-Ground Heat Source and Storage Technology. Recently Canada has become a member of Annex IV, the Heat Pump Centre, which enables Canada to maintain active communication with international research and development. In this connection, a significant development is the formation of a Canadian National Team under the guidance of a National Heat Pump Coordinating Committee. The National Team will be responsible for all of the technical and communication aspects of Canada's interface with the Heat Pump Centre and invites any qualified Canadian national to participate in specific activities, e.g. data gathering and transmittal to the Heat Pump Centre, expert workshops, working annexes and conferences.

In view of the Federal Government's decision to participate financially, Canadian industry now has access to the Heat Pump Centre (HPC) services either:

- directly by communication with the Karlsruhe, West

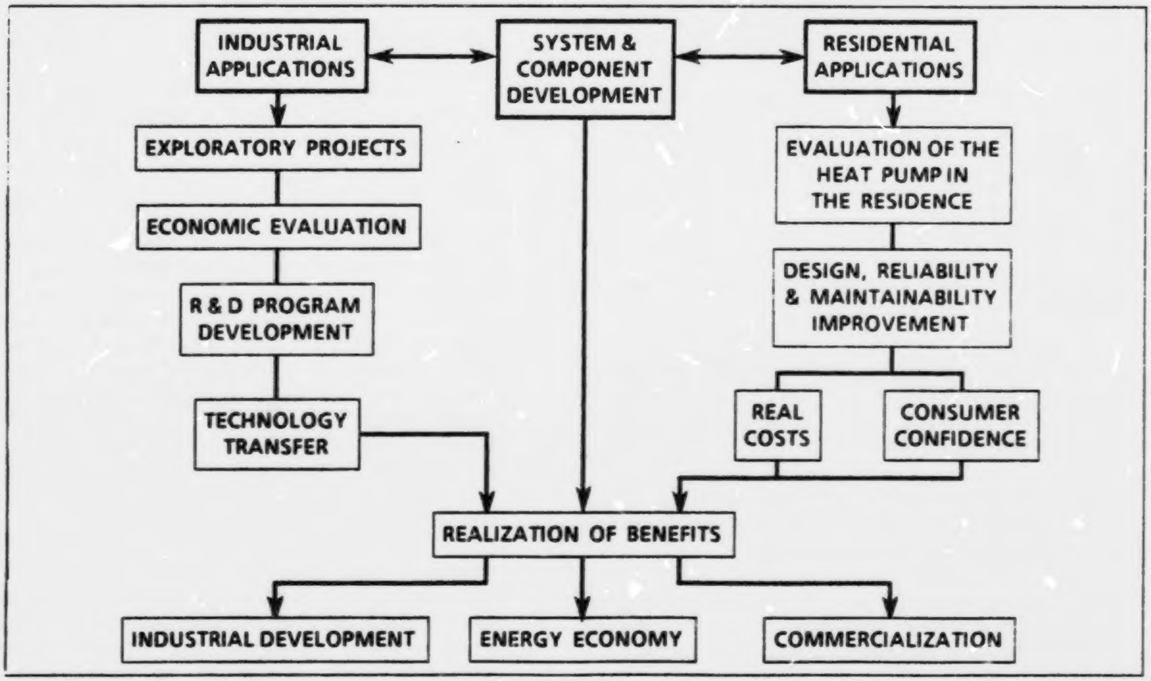
INDUSTRIAL APPLICATIONS
SYSTEM & COMPONENT DEVELOPMENT
RESIDENTIAL APPLICATIONS
EXPLORATORY PROJECTS
ECONOMIC EVALUATION
R & D PROGRAM DEVELOPMENT
TECHNOLOGY TRANSFER
EVALUATION OF THE HEAT PUMP IN THE RESIDENCE
DESIGN, RELIABILITY & MAINTAINABILITY IMPROVEMENT
REAL COSTS
CONSUMER CONFIDENCE
REALIZATION OF BENEFITS
INDUSTRIAL DEVELOPMENT
ENERGY ECONOMY
COMMERCIALIZATION

Figure 1. Program Organization

Germany or Graz, Austria offices;

- through the Canadian National Team;
- through Federal or Provincial Research organizations;
- or through the Canada Institute for Scientific and Technical Information.

Services provided by the IEA through the Heat Pump Centre offices are as follows:

- HPC Newsletter;
- Heat Pump Data Base;
- Expert workshops;
- Conference and Exhibition Representation;
- Engineering Analysis (Graz);
- Business and Commercial Analysis (Karlsruhe);
- Periodic International Conferences.

In the future, working annex activities will be reported both through the HPC Newsletter and through an internal newsletter issued by Energy, Mines and Resources Canada. Participation and communication on annex activities will be welcomed, in the form of comment or criticism or in material contributions. Several examples of possible participation in Annexes current in 1986 are:

- Development of heat pump evaporators;
- High temperature industrial heat pumps;
- Stirling engine heat pumps;
- State and transport properties of high temperature working fluids and non-azeotropic mixtures.

COMMERCIAL RETROSPECT/PROSPECTS

Published information on industry shipments of heat pumps in Canada is not generally available. Through the valuable assistance of a number of industry sources, however, it has been possible to estimate the state of heat pump penetration in Canada between 1975 and 1985.

In 1975, one industry source estimated that 250 split system air-source heat pumps were sold in Canada (split system heat pumps consist of two or more component packages). By 1978, sales had risen to approximately 4000 units. From 1979 onward, shipments of split system and self-contained (packaged) heat pumps, as reported by industry sources, are shown in Figure 2.

The vast majority of split system shipments shown were installed in single family housing in Canada. Cumulative sales of air source heat pumps, based on Figure 2, are approximately 100,000 units. It is estimated, by various sources, that 60,000 of these units are installed in Ontario. According to Statistics Canada (64-202), this represents a 3.5 percent penetration of all single family households with air distribution systems in the province of Ontario; 14.0 percent penetration of all single family households with air conditioning.

The impact on split system shipments of the Canada Oil Substitution program is evident in the large increase in sales between 1980 and 1981 and sustained (with a slight drop during the recession) through 1984. The program was initiated late in 1980 and terminated in March 1985. It illustrates how effective a small grant coupled with contractor/dealer 'push' can significantly advance a product up its technology diffusion curve. Sales may well return to the 1980 level and advance only modestly in the near term in the absence of the contractor/dealer 'push' present during COSP.

Self-contained (packaged) heat pumps experienced virtually no growth during the 1980 through 1984 period. This product class includes rooftop, packaged terminal, and water-to-air heat pumps. These products are largely commercial units, with a small number for well-water or ground source heat pump installations.

In 1985, industry sources estimated that 1400 single package or rooftop air-source heat pumps were shipped in Canada, compared to 800 in 1980. Closed-loop ground-source (water-to-air) heat pumps accounted for approximately 800 units in 1985, up from less than 50 units in 1980. The balance of this category consists of packaged terminal (for use in hotels and motels) and water-to-air heat pumps (for use in office buildings, luxury apartments and condominiums). Industry shipments in 1985 were estimated to be 8600 units. The recent entry of a Canadian manufacturer of packaged terminal heat pumps is to be noted with annual shipments more than doubling in the second year of production. The initial development of this product was supported by the National Research Council. Growth in the self-contained heat pump category is likely, in the near term, due in part to improved economic conditions and the continued growth in closed-loop ground-source application.

Industrial heat pump sales, in Canada, according to one contact, amounted to 15 units in 1985. Little or no growth was foreseen in the immediate future. Further industrial heat pump penetration is unlikely in the absence of incentives for development and demonstration.

TECHNOLOGY OVERVIEW

Heat pump technology and applications are at various stages of development. The following sections briefly describe the current state of the technology. Judgements were derived

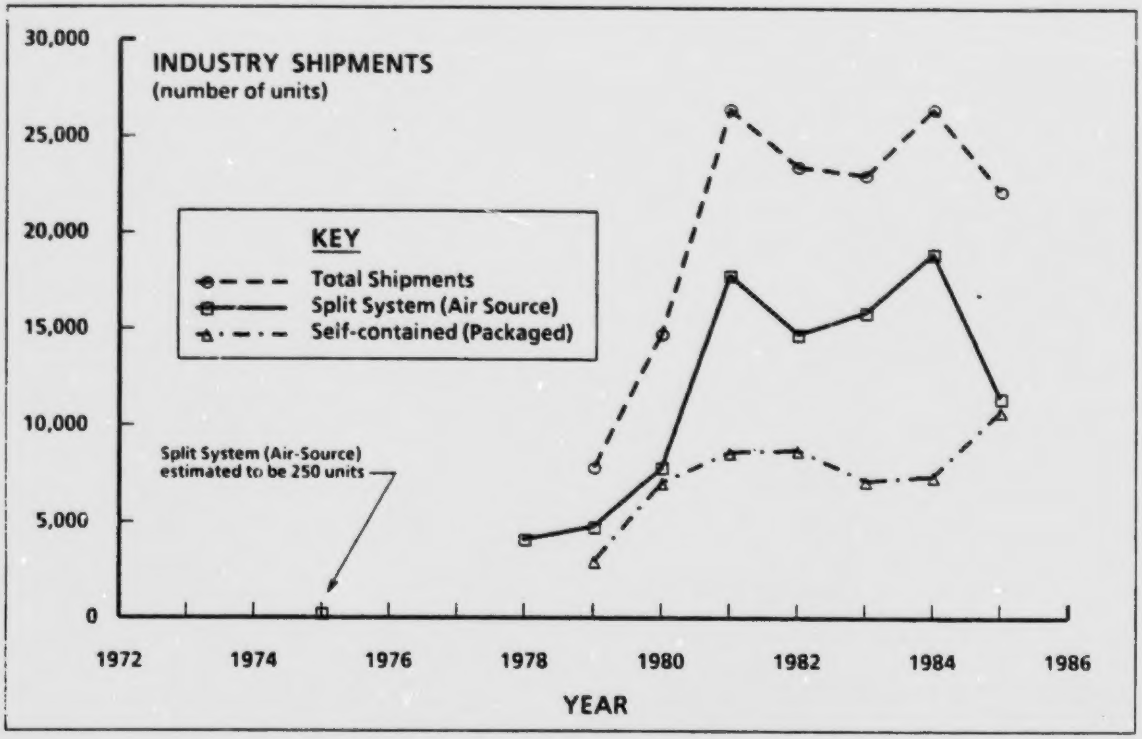

Figure 2. Heat Pump Shipments (All Product Classes) in Canada

from material presented in chapters 2 and 3 as well as from other Canadian organizations and individuals who have an informed opinion on heat pump technology.

Residential Technology

In the residential sector, air-source heat pump technology has matured, with performance and more significantly, reliability, showing dramatic improvement over the past ten years. Northern Climate designs incorporating improved refrigerant control and higher efficiency levels are under development in both Canada and the United States. Current designs, when combined with an electric furnace in new construction, have capital and operating costs competitive with condensing gas furnaces plus electric air conditioning. Air-source heat pumps added to existing oil furnaces offer an attractive payback period, in many cases, without a full capital cost allowance for central air conditioning.

Residential ground or well-water source heat pumps offer significantly higher coefficients of performance than air-source heat pumps. However, with few exceptions, they are still too costly to be attractive in the residential sector even if air conditioning is desired. Ground-source heat pump heat exchanger design, materials and installation techniques have dramatically improved in the past five years.

Residential ventilation heat-recovery heat pump systems are now commercially available in Canada. They may be attractive in building applications where mechanical ventilation is mandatory, as, for example, in houses built to R2000 specifications. The heat extracted by the evaporator from the exhaust air stream is rejected by the condenser to the fresh air stream. Under certain conditions, the evaporator pressure approaches the condenser pressure (low outdoor ambients) and performance is degraded. The addition of a heat exchanger or heat pipe to pre-heat the incoming air and pre-cool the exhaust air can largely overcome this operational difficulty.

Residential hot water heating with a heat pump either in the form of a packaged storage water heater or a desuperheater installed on a central heat pump unit has not yet been demonstrated as a viable option in Canada. The former design extracts heat from the surroundings (basement of house normally) and hence imposes an additional load on the heating system. The latter is relatively unattractive because of the limited hours of operation both in the air conditioning mode and also above the balance point for most Canadian locations.

The performance and reliability of the residential heat pump, in all forms, is no longer a barrier to large-scale market penetration. Central air conditioning sales in Canada doubled between 1983 and 1984 and remained constant in 1985, in contrast to heat pump sales which halved from 1984 to 1985. Promotional efforts by heat pump manufacturers have all but vanished, particularly in Ontario, the largest central air conditioning market. The current widening of the relative cost of natural gas and electricity combined with the incremental cost of $1000 to $1500 for a heat pump over that for central air conditioning are possible reasons for the significant decline in air-source heat pump sales.

Industrial Technology - R114 Closed-Cycle Vapour Compression

The industrial heat pump, in the form of the refrigerant R114 closed-cycle vapour compression cycle, first appeared on the North American market in the late 1970's. The 'Templifier', originally a Westinghouse design, later purchased by McQuay and more recently by the Snyder Corporation in the United States, has been employed in industry for service and process water heating.

A number of R114 industrial heat pump demonstrations, funded by the Division of Energy, have pointed out that the benefits to end users vary over a wide range and are affected by:

- sink and source temperatures;
- cost of equipment;
- cost of maintenance and operation;
- plant operating characteristics and load factor.

Many of the early installations were plagued with equipment breakdowns, which in many cases were a result of plant operating transients which were not foreseen or for which the heat pump was not designed.

More recently, R114 heat pumps in combination with open-cycle steam compressors have been planned to provide medium pressure steam for industrial processes.

Industrial Technology - Chemical Heat Pumps

The Division of Energy undertook to investigate the potential for chemical or absorption heat pumps in distillation processes in Canadian refineries and chemical plants. Absorbents such as lithium bromide, sulphuric acid and sodium hydroxide have been investigated for chemical heat pump use.

Payback periods range from three to eight years, for the applications examined so far, and fall short of the petrochemical industry's typical requirement of two years or less.

The distinct advantage of the chemical heat pump, in the Canadian context, is the potentially high value of goods and services produced in Canada. Canadian companies are acknowledged world leaders in the area of chemical plant and process design, component production and construction.

At the present time there is no chemical heat pump installation in North America. There is, however, a number of firms in the United States offering design and/or installation services. Litwin Industries offers lithium bromide technology (formerly available from Adolph Coors Company); Rocket Research has a laboratory-scale sulphuric acid chemical heat pump under development and Energy Concepts have developed a new chemical heat pump with three absorbents.

In general, the future success of the industrial heat pump can be summarized as follows. In situations where the payback period is two years or less, corporate management will consider approval of an installation provided that confidence is expressed by operations management and the engineering department. Where the payback period is longer, capital resource expenditure for reduced operating costs takes such a low priority that project proposals never come to realization.

In industry, insecurity in energy supply (or economic recession) causes a shortening of the acceptable payback period. On the other hand, a secure energy market, with the attendant low purchased energy costs, lengthens the payback period beyond typical business plan periods of 5 years or more.

It seems possible to conclude that investments by industry in heat pumps require:

- a stable industrial climate;
- stable or predictable purchased energy costs, that are not unrealistically low;
- establishment of energy service (innovative financing) companies to permit risk sharing between implementers and industry.

Commercial Technology

Heat pumps as applied to commercial/institutional buildings are well established.

Closed water-loop unitary water-to-air heat pumps (California cycle) are employed in larger commercial buildings where heating and cooling are often required simultaneously in exterior and interior zones respectively. This equipment is also becoming increasingly popular in luxury high-rise apartments and condominiums.

Unitary rooftop mounted air-source heat pumps account for approximately ten percent of rooftop HVAC unit sales in

Canada. They are generally applied to smaller single-zone commercial buildings where heating and cooling are required at different times of the year. With payback periods of five years or less, rooftop air-source heat pumps are a growth area in Canada in the years to come.

Ground-source heat pumps are becoming increasingly attractive in heating/cooling applications, judging from recent success reported by Earth Systems Limited in the United States. Demonstration projects in this sector are now underway in the province of Ontario.

Another promising area of heat pump application in the commercial sector is the recovery of waste heat from mechanical ventilation exhaust air in high rise apartment/condominium buildings for service water heating. In buildings with central exhaust from suite kitchens and bathrooms, rooftop mounted ventilation units are often in close proximity to penthouse service hot water heating boilers, permitting heat pump service water preheating. At least one Canadian firm offers equipment for this application.

Applications in the commercial sector which have been investigated by the Division of Energy and found not to be cost effective, at this time, include waste water heat recovery in restaurants, laundries and the photofinishing industry. In demonstration projects, the water-to-water heat pump equipment selected experienced operational difficulties due to source/sink temperatures approaching one another, or to process flows exhibiting large fluctuations.

One broad area of heat pump application which offers considerable promise is that of materials or product drying. The Division of Mechanical Engineering examined the technical feasibility of employing heat pumps to dry grains and lumber. The design investigated, in both cases, combined a heat pipe with a conventional vapour-compression heat pump cycle. The heat exchanger precools the drier air, substantially increasing the moisture removal capability of the heat pump evaporator. Energy use, compared to fossil fuel driers, is reduced by 65 to 87 percent. At the same time, drying temperatures are lower; this is conducive to higher quality product.

RECOMMENDATIONS

There remain a number of areas of heat pump technology where further research and development in Canada might be particularly productive:

Residential Heat Pumps

- Canada should continue its research and development efforts in Northern Climate air-source heat pumps. The northern United States potentially represents a large export market for such equipment and no U.S. firm presently has such a unit available. This effort encompasses: optimal refrigerant controls and sensors; flooded evaporator technology; non-azeotropic refrigerant mixtures; improved frost sensing and defrosting control and compressor capacity modulation;
- the exhaust or ventilation air heat-recovery heat pump should be further developed, to overcome problems experienced by early designs, in the anticipation of future growth in superinsulated well-sealed housing;
- efforts to reduce the cost of ground-source heat pump heat exchanger designs and materials and installation techniques should be stepped up, if the ground-source heat pump is to significantly penetrate the single family housing market. A system suitable for retrofit to existing suburban dwellings has not yet been developed.

Industrial Heat Pumps

- high temperature refrigerants such as R113 should be investigated together with refrigerant/oil mixture problems of degradation in the presence of certain materials;
- the production of steam suitable for process plant services using a combination R114 heat pump and steam compressor should be demonstrated;
- development of industrial heat pump computer simulation and optimization tools should be undertaken to avoid many of the pitfalls experienced in early demonstration projects and to provide developers and suppliers of heat pump equipment with an effective way of assessing end-user cost benefit;
- Canada should explore the potential of the sodium hydroxide chemical heat pumps. Work is not currently underway elsewhere and the sodium hydroxide unit can operate at higher temperatures than current lithium bromide designs. As a first step, a laboratory test facility should be constructed, for initial investigations, followed by an in-plant pilot if proven feasible.
- drying processes offer significant energy reduction potential through heat pump application. Further efforts should be undertaken to identify promising areas and to design, develop and demonstrate new heat pump drier technology;

Commercial Heat Pumps

- the cost-benefit of ventilation system heat recovery in large buildings using heat pumps should be demonstrated to commercial building owners and developers.
- a single packaged rooftop heat pump with propane-fired gas furnace back-up rather than traditional electric

resistance heating should be developed. Electrical demand charges would be avoided in most small commercial applications.

Infrastructure

In addition to research and development efforts, issues relating to information, training, consultative services and financing (of large scale industrial heat pumps) need to be addressed. Builders, contractors, developers and industry management need information on the cost-benefit of heat pump applications. Manufacturers need to provide information and/or training to end-users which explains heat pump operating characteristics, maintenance and servicing requirements. Specialist consultants are needed to assist manufacturers and end-users in the application of heat pump technology. Finally, energy service company activities need to be expanded to include heat pump installation, particularly in industry, to overcome the element of risk and to lower payback periods to acceptable levels. Without addressing these areas with the same diligence as the research and development activities recommended above, the future prospects for heat pump technology growth in Canada could be limited.

THIS DOCUMENT

The Review was prepared to provide information on the National Research Council of Canada's research, product development and demonstration experience in the heat pump area. The document will be of interest to researchers in the field, those concerned with commercial application and other readers interested in energy developments.

The document is organized in two sections: Industrial Project Summaries (Chapter 2) and Residential Project Summaries (Chapter 3). The latter is a reporting of the projects undertaken by the Divisions of Building Research and Mechanical Engineering. Projects planned and managed under the Division of Energy's Heat Pump Program are summarized in Chapter 2.

The Project Index, Chapter 4, provides the reader with additional information regarding the author and/or external contractor, scientific authority, contract serial number and year of completion. This information will allow the more serious reader to obtain a copy of the original report or NRCC publication by writing to:

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National Research Council Canada
Ottawa, Ontario
K1A OR6

In closing, it is intended that the research and industrial community at large finds this Review of Heat Pump Research and Development Activities to be both technically informative and commercially rewarding.

CHEMICAL HEAT PUMPS IN CANADIAN INDUSTRY

Acres Consulting Services Ltd., Toronto, performed this market study between January and March 1984.

PROJECT RATIONALE

Moderate progress has been made in developing new energy cascading concepts using chemical heat pumps (CHP). It is anticipated that CHP technologies and applications will be slowly undergoing a period of refinement from a pure research and development stage to commercialization.

The objectives of this study were:

- to provide an overview of CHP systems;
- to identify a specific industrial application for a CHP to apply to steam generation on high quality heat recovery;
- to estimate costs and benefits expected in the application;
- to assess the scope for the proposed application on a industry-wide basis.

OVERVIEW OF CHP SYSTEMS

The most common chemical or absorption refrigeration units are used in air conditioning applications. When their primary purpose is to provide heating rather than refrigeration, these units are classified as CHP's.

Ammonia and lithium bromide CHP's are best suited for applications with maximum condenser temperatures of about 100 C, although higher temperatures are possible. Sulphuric acid and sodium hydroxide CHP's are most suitable for temperatures approaching 150 C. Higher temperature CHP's such as metal hydride or zeolite CHP's have not been well researched.

CHP's IN PAPER MANUFACTURING

Paper manufacturing, using thermo-mechanical pulp refining (TMP), provides large flows of waste heat. The feasibility of utilizing the waste heat from the TMP refiner to dry paper was considered.

The Ontario Paper Co. at Thorold was approached with an idea for a sulphuric acid or sodium hydroxide CHP to upgrade TMP waste heat for paper drying. Their operations had been recently expanded and modernized, and they were already evaluating a proposal to upgrade clean steam from 260 kPa to 520 kPa using a thermocompressor.

Principal mass flows were calculated for a CHP using either sulphuric acid or sodium hydroxide as the absorbent. A COP of 1.5 was predicted. The economics of the system are discussed in Figure 1. It was found, however, that the relatively simple, proven thermocompressor heat pump would be the more practical alternative for upgrading steam to 520 kPa.

CHP's IN PAPER DRYING

A drying application was selected as an alternative to the steam upgrading example. An open cycle sodium hydroxide CHP was examined as described in Figure 2. Waste steam, such as from an atmospheric pressure TMP refiner, could be used as a heating source in the generator. Moist air from a paper dryer would be dried and heated as it is mixed with concentrated sodium hydroxide in a contacting tower (absorber). The economics of this alternate CHP configuration were not explored.

CHP POTENTIAL IN CANADIAN INDUSTRY

CHP application was considered limited to heat streams between 32 C and 290 C. It was determined that the maximum technical potential for energy recovery by CHP's in Canadian industry was 192 PJ (1978).

Only a certain portion of the maximum recoverable energy could be recovered by CHP's in practice. It was assumed that 10 percent of the recoverable energy in the 100 C to 290 C range; 20 percent in the 50 C to 100 C range; and 100 percent below 50 C could actually be recovered by CHP's. This resulted in a total technical potential for CHP's in Canadian industry of 55 PJ (1978).

CONCLUSIONS AND RECOMMENDATIONS

Chemical heat pump applications are relatively unknown to industrial users. However, their application is technically feasible and attractive when no other potential uses for waste energy are available or when rejected heat energy is at a low temperature. Sulphuric acid and sodium hydroxide CHP's are well suited for providing 150 C process steam.

The payback period for a 4.4 MW sulphuric acid or sodium hydroxide CHP would be 8.3 years - higher than industry normally accepts. This could be improved with further development.

It was recommended that a pilot project be implemented to forward CHP development and application.

UPDATE

The open cycle CHP for paper drying described in Figure 2 was explored in detail by the consultant in a subsequent study (see page 2-21).

DATA:

Process:	Steam generation
Waste heat source:	Waste steam 1.5 MW
Generator input energy:	3.0 MW
Heat pump output:	4.5 MW
Installed cost:	$1.3 million
COP:	1.5
Payback:	8.3 years

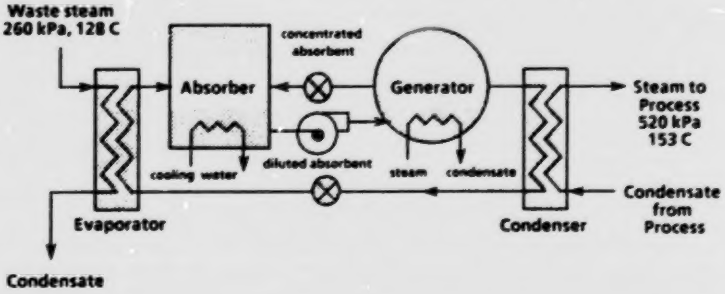

Waste steam
260 kPa, 128 C
Absorber
concentrated absorbent
Generator
diluted absorbent
cooling water
steam
condensate
Steam to Process 520 kPa 153 C
Condensate from Process
Evaporator
Condenser
Condensate

Figure 1. Illustration of the Economics of a Sulphuric Acid CHP for Steam Production

The chemical heat pump shown in Figure 1 could use either sulphuric acid or sodium hydroxide as the absorbent. Low grade steam (260 kPa) serves as an ideal waste heat stream. Liquid water in the chemical heat pump is vaporized in the evaporator at a rate of 1.5 MW. This water vapour is absorbed by concentrated sulphuric acid (or sodium hydroxide). The sulphuric acid, now diluted, is pumped to the generator where the water is revaporized. Energy for this vaporization comes from high pressure steam or other heat source. Finally, the water vapour is condensed, releasing energy at the rate of 4.5 MW at the condenser; 520 kPa (75 psig) steam is produced. The system COP is 1.5. Assuming a boiler efficiency of 71 percent and mechanical and piping losses of 10 percent, the net energy savings of 1.5 MW is equivalent to 2.4 MW of boiler fuel.

Based on a total capital cost of $1,300,000 including all hazard control and environmental protection costs and an average annual energy savings of $340,000 (equivalent natural gas heating at $0.16/m^3), a discounted payback of 8.3 years is predicted. A return on investment of 16% is also predicted. An implementation schedule is estimated to be 40 months.

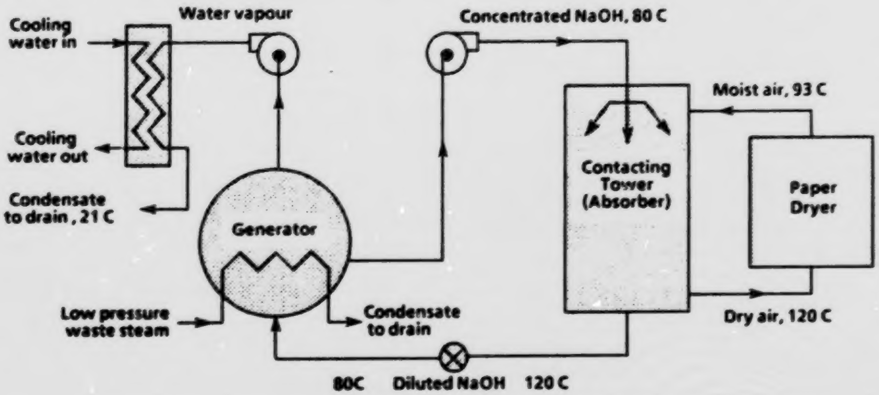

Cooling water in
Water vapour
Concentrated NaOH, 80 C
Moist air, 93 C
Cooling water out
Contacting Tower (Absorber)
Paper Dryer
Condensate to drain, 21 C
Generator
Low pressure waste steam
Condensate to drain
Dry air, 120 C
80C
Diluted NaOH
120 C

Figure 2. Open Cycle NaOH Absorption Chemical Heat Pump for Paper Drying

Warm moist air from the paper drying equipment enters the top of the contacting tower. Simultaneously, sodium hydroxide solution is sprayed into the top of the tower, mixing with the moist air stream. A exothermic chemical reaction occurs as the air/sodium hydroxide mixture passes down through the tower, with water transfering from the air to the sodium hydroxide. The air, now warmer and dryer, returns to the paper drying equipment via a de-mister which removes any carried-over sodium hydroxide.

The diluted sodium hydroxide is fed to a generator for regeneration (i.e. removal of the moisture picked up in the contacting tower) and is subsequently recycled to the tower. The water vapour driven off is condensed using cooling water.

DISTILLATION INDUSTRIES

Stone and Webster Canada Ltd., Toronto, performed the following chemical heat pump market study. The report was published in March 1984.

PROJECT RATIONALE

Chemical heat pumps are ideally suited for fractional distillation processes. Characteristically, a fractional distillation process requires a large supply of energy (that is, at elevated temperatures) but only consumes a small fraction of the supplied energy. Thus, it is a prime candidate for energy conservation measures. Chemical heat pumps can conserve up to 50 percent of the required distillation energy. Industries which use distillation processes extensively are refineries, petrochemical plants, chemical plants and distilleries.

The objectives of the study were:

- to identify at least two specific industrial applications for distillation process heat recovery using chemical heat pumps and to determine energy conservation potential for each;
- to determine the scope of a fully detailed feasibility study, including costs, benefits and schedule for the design and completion of a demonstration project;
- to estimate the industry-wide applicability and benefits of the proposed chemical heat pumps.

INDUSTRIAL APPLICATIONS

After communications with many corporations, 30 distillation processes were documented. Documentation included all relevant temperature and flow rates to and from the distillation columns. An analysis of the data provided the energy recovery potential of using chemical heat pumps (CHP's) for each.

Two of these distillation processes were examined in greater detail to determine the economic merits of applying CHP's. Lithium bromide CHP's were chosen for the evaluation, as their use is well proven in other industrial applications. One distillation process involved a forward cycle CHP for distillation of isobutane, while the other involved a reverse cycle CHP for distillation of water from a butanol stream. These detailed in Figures 1 and 2 respectively. Each was predicted to have an attractive payback of about 2 years.

DETAILED FEASIBILITY STUDY SCOPE

The technology required for the successful application of CHP's in distillation processes was found to be available, based on the analysis of the two processes described above and the knowledge of a similar plant application in Japan. Therefore it was recommended the next step toward commercialization would be the preparation of a detailed engineering design and a schedule for the activities, from design through to construction and commissioning. The cost of undertaking a detailed engineering design was estimated to be $35,000. A schedule was prepared, indicating a period of 15 months from the inception of an engineering design study to the completion of commissioning.

MARKET STUDY

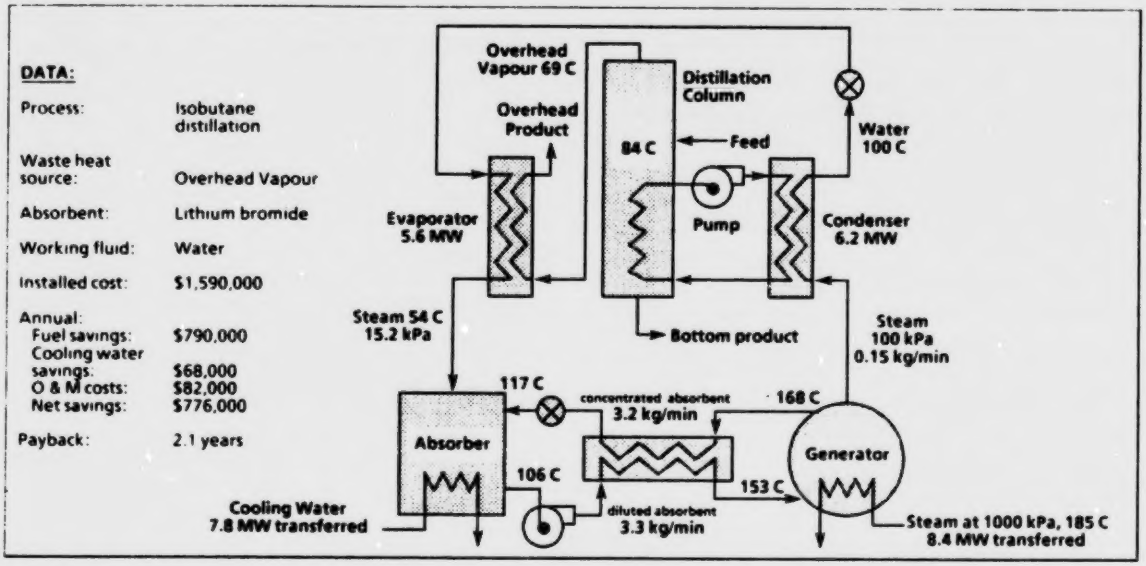
The total distillation energy use in the Canadian petrochemical industry was estimated to be 171 PJ per year. After a detailed analysis of process temperatures and capacities of all of the distillation processes used in the industry, processes using a total of 34.5 PJ per year were found to be technically suitable for CHP applications. Assuming CHP's would be competitive with alternate heat recovery options in only 20 percent of these processes, and that CHP's would recover 45 percent of the process energy, the energy recovery potential for CHP's in Canada was predicted to be 3.1 PJ per year.

CONCLUSIONS AND RECOMMENDATIONS

Two CHP applications were studied in detail, resulting in preliminary cost and benefit estimates. The first, for recovering 5.6 MW using a forward cycle CHP for isobutane distillation, had a predicted payback period of 2.1 years. The second, for recovering 3.0 MW using a reverse cycle CHP for butanol dewatering, had a predicted payback period of 2.2 years.

Estimated capital cost of all applications reviewed ranged from $220,000 to $3,000,000 with paybacks in the order of 1 to 3 years. The total market potential for energy recovery using CHP's in Canadian industrial distillation processes was estimated at 3.1 PJ per year, equal to $18.5 million per year of equivalent oil consumption (1983).

Considering the wide interest of the industries surveyed and the significant potential for Canadian market penetration, it was recommended that a detailed engineering design be undertaken to refine CHP economics.



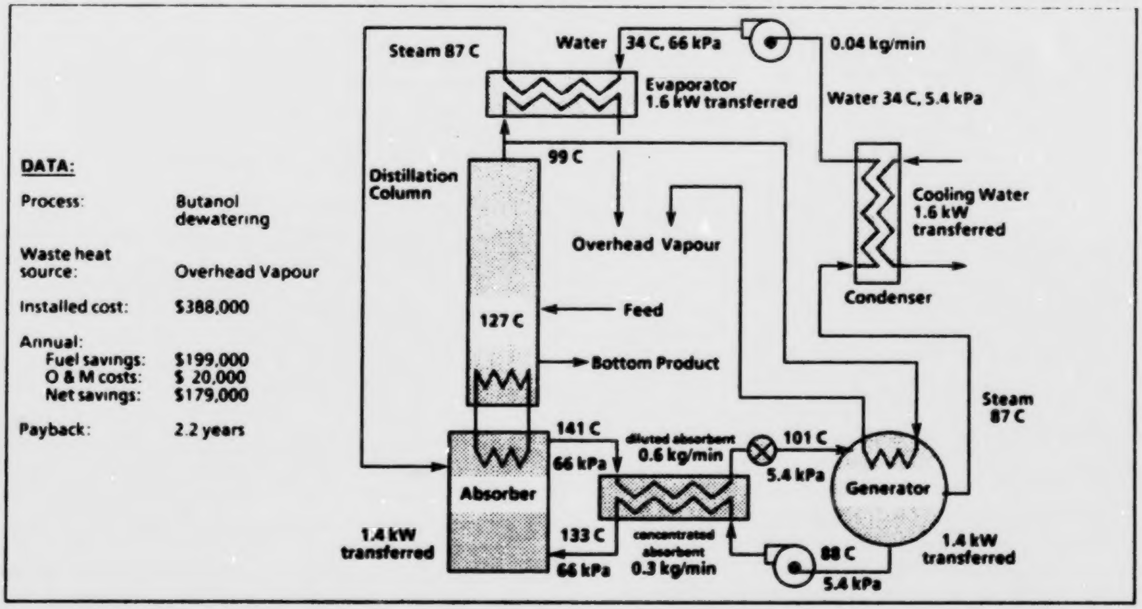
DATA:

Process:	Isobutane distillation
Waste heat source:	Overhead Vapour
Absorbent:	Lithium bromide
Working fluid:	Water
Installed cost:	$1,590,000
Annual:	
Fuel savings:	$790,000
Cooling water savings:	$68,000
O & M costs:	$82,000
Net savings:	$776,000
Payback:	2.1 years

Overhead
Vapour 69 C
Distillation
Column
Overhead
Product
84 C
Feed
Water
100 C
Evaporator
5.6 MW
Pump
Condenser
6.2 MW
Steam 54 C
15.2 kPa
Bottom product
Steam
100 kPa
0.15 kg/min
117 C
concentrated absorbent
3.2 kg/min
168 C
Absorber
Generator
106 C
153 C
Cooling Water
7.8 MW transferred
diluted absorbent
3.3 kg/min
Steam at 1000 kPa, 185 C
8.4 MW transferred

Figure 1. Forward Cycle LiBr Absorption Heat Pump for a Distillation Process.

In this example isobutane is being produced through a distillation process. Heat is recovered from the isobutane at the evaporator. Heat produced by the CHP drives the distillation process.



DATA:

Process:	Butanol dewatering
Waste heat source:	Overhead Vapour
Installed cost:	$388,000
Annual:	
Fuel savings:	$199,000
O & M costs:	$ 20,000
Net savings:	$179,000
Payback:	2.2 years

Steam 87 C
Water
34 C, 66 kPa
0.04 kg/min
Evaporator
1.6 kW transferred
Water 34 C, 5.4 kPa
99 C
Distillation
Column
Cooling Water
1.6 kW
transferred
Overhead Vapour
Condenser
127 C
Feed
Bottom Product
Steam
87 C
141 C
diluted absorbent
0.6 kg/min
101 C
66 kPa
5.4 kPa
Absorber
Generator
1.4 kW
transferred
133 C
concentrated
absorbent
0.3 kg/min
88 C
66 kPa
5.4 kPa
1.4 kW
transferred

Figure 2. Reverse Cycle LiBr Absorption Heat Pump for a Distillation Process.

In this example water is removed (distilled) from an isobutane feed. Waste heat from the distilled water vapour is recovered at the evaporator and the generator. The Chemical Heat Pump upgrades this heat, where it is made available at the absorber. Heat provided at the absorber is used to drive the distillation process. In contrast to a forward cycle design, the coldest point in the heat pump circuit is at the condenser, where a coolant is required to remove heat.

DISTILLATION PROCESS - BUTANOL DEWATERING

Stone and Webster Canada Ltd. of Toronto, performed this feasibility study, dated January 1986.

PROJECT RATIONALE

The consultant had prepared an initial market study (see page 2-3) which found chemical heat pumps to have strong potential for application in distillation processes such as those found in the petroleum and petrochemical industries. Payback periods on the order of 2 years were predicted.

The objectives of this project were to:

- identify a suitable industrial application;
- specify a chemical heat pump package; and
- to prepare a commitment level capital cost estimate, economic analysis, and schedule.

SITE SELECTION AND DESIGN CRITERIA

BASF Canada Limited of Laval, Québec. and Celanese Canada of Edmonton Alberta, were identified as potential candidates for installation of a CHP system. Each had a number of distillation columns with suitable heat recovery potential.

After consultations and site visits, a butanol dewatering column at the BASF facility was determined to be the most suitable application. Waste heat recovered from water vapour distilled from a butanol stream would be applied to the reboiling of butanol in the distillation column. Initial calculations had indicated a 48 percent heat recovery using a reverse cycle chemical heat pump system. Relevant process flow rates, temperatures and compositions were confirmed. It was pointed out that materials containing copper could not be used in the design where they would come in contact with the process fluids.

CHP operating principles are discussed in detail in the initial market assessment study (page 2-1). It should be noted that reductions of up to 50 percent in the energy required by the reboiler are possible by incorporating a CHP into a distillation process. In contrast to vapour compression heat pumps, this savings is derived without requiring a substantial input of mechanical power.

The selection of the reverse cycle CHP was aided by a computer program. A schematic of the operation is shown in Figure 1.

DETAILED DESIGN AND ECONOMICS

As design work proceeded it became apparent that certain data previously provided was incorrect, which invalidated original assumptions about the installation site at BASF. Initial evaluation revealed a 48 percent reduction in reboiler heat requirement. Evaluation of the new data revealed only a 31 percent reduction in reboiler heat requirement. The chief discrepancy was overhead vapour duty. Initial data, it turned out, described the design output of an existing overhead air cooled condenser. After subsequent investigation it was found that the actual overhead duty was significantly less. The data discrepancies are shown in Table 1. It was also revealed that the distillation column operated for only five months of the year.

Table 1. Differences Between Initial Information and Actual Data

	Initial Information	Detailed Data Analysis
Overhead Duty (MW)	2.99	2.20
Overhead Temperature (C)	99.0	91.0
Reboiler Duty (MW)	2.93	2.29
Reboiler Temperature (C)	127	120
Reverse Cycle CHP Reboiler Duty Reduction	48%	31%

Specifications were prepared for the CHP design and bids requested. Only one quotation was received, for an installed cost of $1,085,000. This did not compare favourably with an anticipated cost of about $650,000, based on information received in the previous market study. The primary reason for the increased cost was the need to use titanium tubing in the system heat exchangers rather than standard copper/nickel tubing material.

Two factors, the decreased heat recovery and increased cost, altered the scenario to the point where the initial 2 year payback period for this installation would in fact be approximately eight years. For this reason the location was rejected as being unsuitable for CHP application.

CONCLUSIONS

It was decided after further investigation that the application was not economically feasible, and that

DATA:

Process:	Butanol dewatering
Waste heat source:	Overhead vapour
Installed cost:	$1,085,000
Payback:	8 years

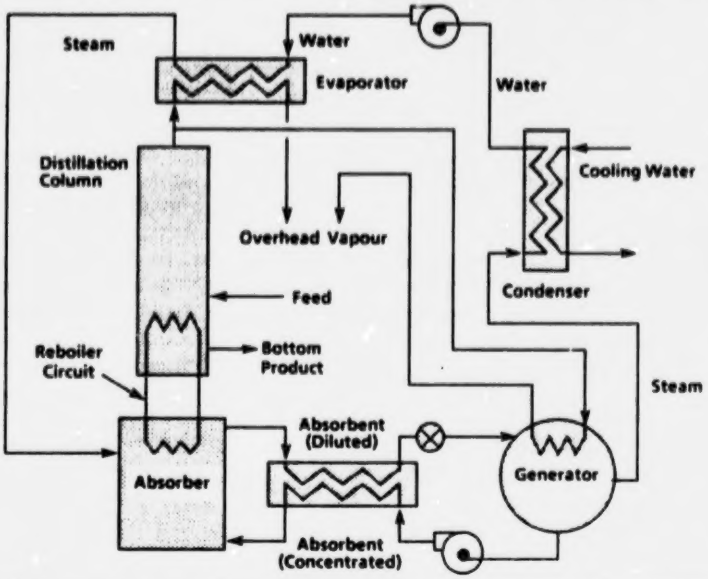


Figure 1. Reverse Cycle LiBr Absorption Heat Pump for a Distillation Process at BASF of Laval, P.Q.

In this example water is removed (distilled) from a butanol feed. Waste heat from the distilled water vapour is recovered at the evaporator and the generator. The Chemical Heat Pump upgrades this heat, where it is made available at the absorber. Heat provided at the absorber is used to drive the distillation process.

the project would not be continued. It should be kept in mind that this application is not indicative of most, and that it failed due to deviations from initial information used in making the economic analysis.

Following these findings, it was decided that new potential prospects be evaluated. The other columns at BASF, as well as the columns at Celanese Canada, were reinvestigated but subsequently rejected. Six other companies were contacted and potential applications were analyzed. No suitable distillation columns were located.

The results of the plant surveys began to indicate that existing plants were not suited to CHP installation. Retrofit of such a system proved to be uneconomical for one more of the following reasons:

- process fluids were incompatible with copper alloys normally used in heat exchangers, necessitating the use of alloys such as titanium and stainless steel;
- heat available from the overhead condenser was insufficient to provide an adequate reduction in reboiler heat required;
- the temperature lift between the bottom and top of the distillation column was too great;
- space limitations;
- modifications to the utility supply were required, which complicated the installation.

In general, applications can be economical in instances where the available overhead heat is 1.5 MW or greater. An application in which titanium is required in the heat exchanger design should be avoided due to the increased capital cost. However, certain applications in which reboiler heat reductions and energy costs are high may prove to be economical.

RECOMMENDATION

It was recommended that further consideration be given to implementation of the absorption heat pump systems, as the companies surveyed showed much interest in such systems. Canadian content of goods and services would be high in such applications.

REFINERY - TOLUENE DISTILLATION

Stearns Catalytic Ltd., of Sarnia, Ontario, carried out this chemical heat pump feasibility study between January and March 1984.

INTRODUCTION

The operation of a distillation tower requires that the latent heat of evaporation be removed from the overhead vapour. This latent heat is usually discarded through an air or water cooled condenser. As well as being capable of recovering approximately 45 percent of this heat and producing steam or other heating medium, a chemical (absorption) heat pump has:

- very little power requirement;
- low operating costs;
- good mechanical reliability, as there are no moving parts other than pumps;
- little technical sophistication, as the equipment mainly consists of heat exchangers; and
- excellent turndown on reduced load.

The objective of this work was to provide a detailed economic analysis of installing a chemical heat pump (CHP) to produce steam, using waste heat from a refinery distillation process.

DESIGN

Sunoco Incorporated operates a number of distillation columns at its refinery in Sarnia, Ontario. Sunoco was approached and expressed interest in CHP technology for waste heat recovery. Of the distillation processes in operation at the refinery, the toluene process was chosen because of its high overhead condenser duty (7.4 MW) and high temperature (127 C).

A general constraint was imposed on the design from the start: the heat pump should have little or no interference with the process operation. Should there develop a problem with the heat pump prototype, the process could continue to operate in its original manner without loss of toluene production. The CHP would be designed to produce medium pressure steam, displacing some of the refinery's purchased steam.

The design chosen was submitted by Adolph Coors Company (Colorado). Coors was licensed in 1983 to design, fabricate and market lithium bromide CHP's developed by the Battelle Memorial Institute (Ohio). Based on process flows provided by the consultant, Coors provided two CHP designs. Each was a variation of a reverse cycle CHP configuration. The waste heat recovery rate would be the same with either configuration. They are shown in Figure 1.

ECONOMICS

Specifications were prepared for the new equipment and the piping tie-ins with the existing plant. To get an accurate estimate for the capital cost of the

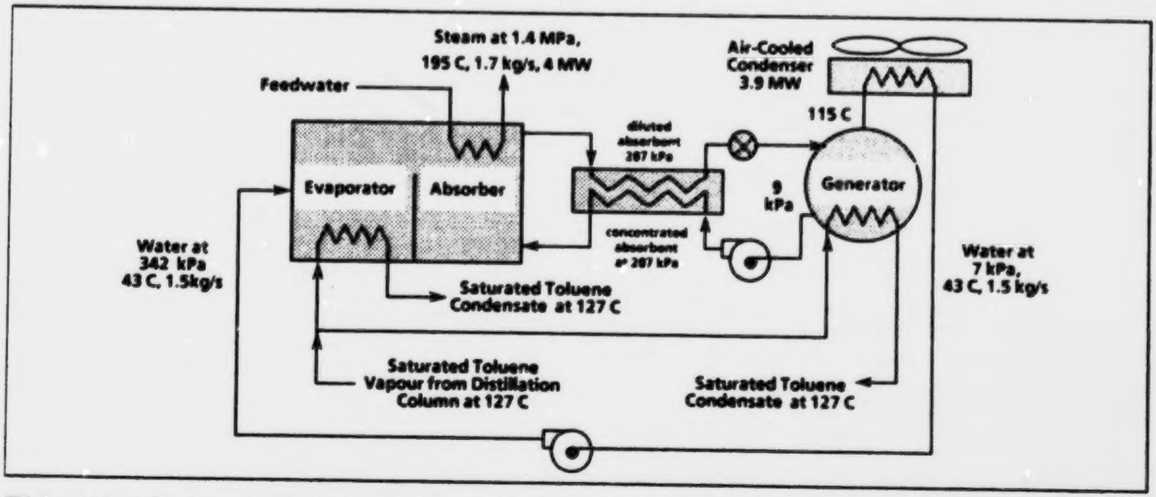

Steam at 1.4 MPa,
195 C, 1.7 kg/s, 4 MW
Feedwater
Air-Cooled
Condenser
3.9 MW
115 C
diluted
absorbent
207 kPa
Evaporator
Absorber
9
kPa
Generator
concentrated
absorbent
at 207 kPa
Water at
342 kPa
43 C, 1.5kg/s
Water at
7 kPa,
43 C, 1.5 kg/s
Saturated Toluene
Condensate at 127 C
Saturated Toluene
Vapour from Distillation
Column at 127 C
Saturated Toluene
Condensate at 127 C

Figure 1a. Reverse Cycle LiBr Chemical Heat Pump for Steam Production (CASE 1).

investment, all equipment and material costs were specified and firm prices were obtained from vendors. The resultant capital cost of each was predicted to be approximately $2,310,000, and within an accuracy of ± 10 percent.

The design calls for a production of 1.7 kg/s of steam at 195 C (see Figure 1). Until a detailed engineering design is undertaken, only 85 percent of this production, or 1.4 kg/s of steam, was assumed available. This steam was calculated to displace $675,000 worth of purchased steam per year, with a net savings of $622,000 after accounting for $13,000 in annual operating costs. A payback period of 3.5 years results ($2,310,000 ÷ $622,000 per year).

Sunoco was looking for a payback period of less than 2 years or a return on investment of over 60% on an energy conservation project at the time, and therefore found the proposed CHP unattractive.

MARKET

Coors had been surveying companies for possible CHP applications for a year, and their results were considered the best available without an extensive market survey. They found that there were an average of 2.5 distillation columns per petroleum or petrochemical plant suitable (i.e. columns with 3 MW to 6 MW of waste heat) for CHP waste heat recovery. Only 10 percent of the columns were estimated to be practical for CHP applications after considering limitations such as available space, temperature boost required and economics (payback periods less than 3 years).

Based on 101 Canadian plants (1983), there were an estimated 25 suitable applications for CHP's in Canada (2.5 columns/plant x 101 plants x 10% practical sites).

Based on CHP capability of recovering 45 percent of available waste energy, the total Canadian heat recovery potential of CHP technology in the petroleum and petrochemical industries is:

4.5 MW(avg.)/column x 25 columns x

45% CHP recovery = 50 MW

CONCLUSIONS

Payback periods between 1 and 3 years for CHP's should generally be attractive to industry. Based on the payback period of 3.5 years predicted for the toluene distillation heat pump application, it was concluded that industry would not find CHP's attractive at present fuel prices.

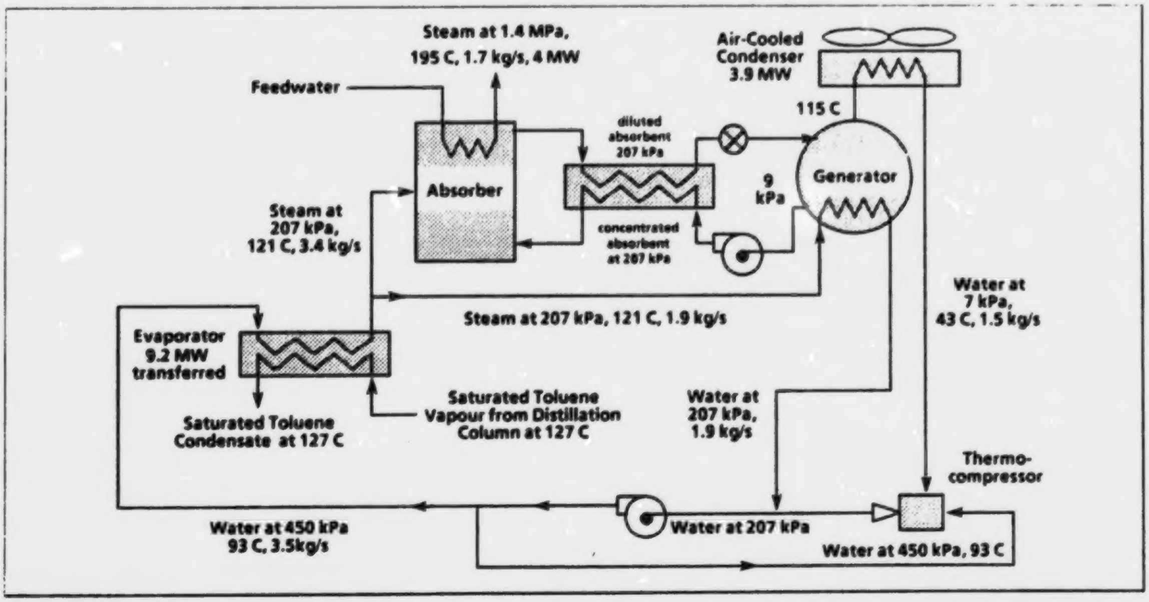

Steam at 1.4 MPa,
195 C, 1.7 kg/s, 4 MW
Air-Cooled
Condenser
3.9 MW
Feedwater
115 C
diluted
absorbent
207 kPa
Absorber
9
kPa
Generator
Steam at
207 kPa,
121 C, 3.4 kg/s
concentrated
absorbent
at 207 kPa
Water at
7 kPa,
43 C, 1.5 kg/s
Steam at 207 kPa, 121 C, 1.9 kg/s
Evaporator
9.2 MW
transferred
Saturated Toluene
Vapour from Distillation
Column at 127 C
Water at
207 kPa,
1.9 kg/s
Saturated Toluene
Condensate at 127 C
Thermo-
compressor
Water at 207 kPa
Water at 450 kPa
93 C, 3.5kg/s
Water at 450 kPa, 93 C

Figure 1b. Reverse Cycle LiBr Chemical Heat Pump for Steam Production (CASE 2).

PULP AND PAPER INDUSTRY - STEAM GENERATION

G.A. Robb Associates of Ottawa performed this pulp and paper industry heat pump feasibility study between June and December, 1983.

PROJECT RATIONALE

The pulp and paper industry uses three times more oil and as much natural gas as any other industry in Canada and has an abundance of low temperature (65 C to 82 C) waste heat sources. In the production of newsprint, paper is typically dried with medium pressure steam (between two and three atmospheres). Conventional refrigerant cycle heat pumps can be used to recover some of this waste heat to produce low pressure steam, which can in turn be upgraded to medium pressure process steam using steam compression heat pumps.

The objectives of this study were:

- to determine a vapour compression heat pump design for producing medium pressure steam;
- to prepare an economic assessment of the design;
- to estimate the market potential for this heat pump in the pulp and paper industry.

SYSTEM DESIGN

The process conditions at Consolidated Bathurst's Port Alfred Newsprint mill in La Baie, Québec were evaluated for their waste heat recovery potential. The wastewater stream from a pulping machine would provide a suitable heat source for waste heat recovery. An R114 closed cycle heat pump could be used to produce low pressure steam (about 70 kPa). This low pressure steam would be upgraded using an open cycle steam compressor to produce 240 kPa steam for paper drying. The proposed system is shown in Figure 1.

The Consolidated Bathurst mill had an existing heat recovery system, capable of supplying clean water to the evaporator of the R114 heat pump. This avoided fouling problems which would have occurred had contaminated process wastewater been used directly. The evaporator was designed to recover 3.2 MW of energy from the wastewater, cooling it from 72 C to 65.5 C.

Low pressure steam (69 kPa, 90 C) would be produced at the condenser. This would be compressed to 240 kPa using a steam centrifugal compressor, to provide steam at a rate of 4.7 MW for paper drying.

New developments in screw and centrifugal steam compressors have reduced the cost of open cycle heat pumps. In many cases, the costs are only a half or a third of previous levels. Choice of the compressor for this installation should be made after careful evaluation during a detailed engineering design.

Two alternatives to the above system were considered. A larger scale R114 heat pump could be engineered to produce medium pressure process steam directly. However, for reasons of economics it would have to it have greater capacity than units having more general application. As well, such a system was about to be demonstrated in Sweden, and it would be prudent to observe it's success. The second alternative was an R114 heat pump coupled with a methanol heat pump for process steam production. This was under development by Westinghouse, and its performance too should be observed.

ECONOMICS

Steam was generated using oil-fired boilers, and after accounting for efficiency losses, the cost was $4.74/GJ. Electricity was 1.7¢/kWh. Based on these costs a net savings of $710,000 per year was estimated. With a total capital cost estimated to be $1,320,000, the system would have a 1.9 year payback period. A discounted cash flow analysis revealed an internal rate of return of 24 percent.

Most of the potential for this application is in Québec, Ontario and British Columbia where electrical rates are relatively low. A study recently completed by G.A. Robb Associates for the Canadian Electrical Association concluded that this technology could displace 5.4 million barrels of oil equivalent, valued at roughly $160,000,000 per year in the newsprint industry alone.

CONCLUSIONS

A steam generating R114 heat pump coupled with a steam compression heat pump capable of producing 4.7 MW of medium pressure steam for a paper mill was estimated to cost $1.3 million and have a 2 year payback period.

RECOMMENDATIONS

It was recommended that a skid mounted R114 heat pump be designed. Once available, this heat pump together with a steam compressor should be demonstrated in an industrial installation, such as Consolidated Bathurst's Port Alfred mill.

DATA:

Process:	Pulp and paper production
Waste heat source:	White water
System output:	4.7 MW
COP:	2.7
Installation cost:	$1,320,000
Savings:	$710,000/year
Payback:	1.9 years
I.R.R.:	24%

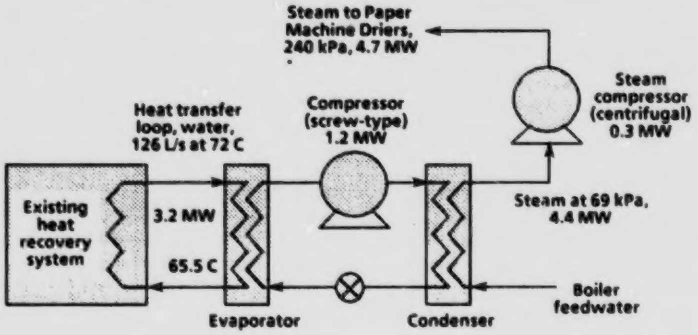

Steam to Paper Machine Driers, 240 kPa, 4.7 MW
Steam compressor (centrifugal) 0.3 MW
Heat transfer loop, water, 126 L/s at 72 C
Compressor (screw-type) 1.2 MW
Existing heat recovery system
3.2 MW
65.5 C
Steam at 69 kPa, 4.4 MW
Boiler feedwater
Evaporator
Condenser

Figure 1. Proposed Heat Recovery from Consolidated Bathurst's Port Alfred Newsprint Mill in La Baie, Québec, Using a Vapour Compression Heat Pump System.

The wastewater stream from a pulping machine provides the heat source. The Consolidated Bathurst mill has an existing heat recovery system, capable of supplying clean water to the evaporator of the R114 heat pump (shown above). The R114 heat pump produces low pressure steam, which is upgraded using an open cycle steam compression heat pump, to produce 240 kPa steam for paper drying.

DESIGN OF A STEAM GENERATING HEAT PUMP

This design study was undertaken by J.H. Lock and Sons Limited of Dartmouth, Nova Scotia, dated December 1985.

PROJECT RATIONALE

The cost - effective production of steam with a heat pump would be desirable to a number of process industries. For example in a pulp and paper mill, there is a constant requirement for steam - but not for hot water, the common industrial heat pump output. Hot water is already in abundance in the form of steam condensate.

The objective of this study was to prepare a design and cost estimate for a steam generating heat pump.

SYSTEM DESIGN

Design constraints were based on the process requirements of a pulp and paper plant in Grand' Mère, Québec operated by Consolidated Bathurst Incorporated. The basic design would have application across the pulp and paper industry, as well as to other industries which use large quantities of process steam. Waste heat would be recovered at a rate of 3.1 MW from 68 C white water from a pulping machine using an R114 closed cycle heat pump. The R114 heat pump would generate low pressure steam to be upgraded to medium pressure steam (340 kPa) using an open cycle steam compression heat pump, as shown in Figure 1.

The evaporator of the R114 heat pump is a single pass shell and tube heat exchanger with stainless steel on the tube side (white water side). Previous experience by Consolidated Bathurst indicates that fouling of the tubes with wood fibre from the white water is minimized if flows through the tubes are in excess of 1.7 m/s. The design rate for the evaporator is 2.5 m/s.

A liquid/suction heat exchanger is used to cool the saturated liquid refrigerant exiting the condenser while superheating the refrigerant vapour from the evaporator. Unless the refrigerant vapour is superheated, some refrigerant would condense during compression, and liquid refrigerant could damage the compressor.

A screw compressor rather than a centrifugal compressor is best suited for the R114 circuit because:

- there is a positive means of capacity reduction with almost proportional power savings; and
- less expensive (half the cost in the desired capacity range).

The condenser (steam generator), like the evaporator, is of shell and tube design. Refrigerant condenses on the outside of the finned tubes at 110 C. Carbon steel is currently specified for the condenser. Since oil and refrigerant may decompose (at an annual rate of 2 percent) in the presence of carbon steel at the high temperatures found in the condenser, stainless steel (an annual decomposition rate of 0.01 percent) may be the preferred option.

Water from a circulating pump enters for a single pass through the condenser tubes. A saturated mixture comprised of four parts water and one part steam exits the tubes at 105 C. The exiting saturated water, collected in a receiver, is recirculated through the condenser while the saturated steam is drawn off by a steam compressor. Treated feedwater is added to the receiver in order to balance the steam production flowrate.

Low pressure steam (120 kPa) drawn by the steam compressor would be compressed to 340 kPa. Two steam compressor types were investigated; one a screw compressor, the other a two-stage centrifugal compressor. The centrifugal compressor, although the least expensive of the two options, is subject to relatively high power consumption under low load conditions. In contrast, the screw compressor consumes power in proportion to load. It may be the least expensive option in the long run because of lower operating costs in operations where waste heat sources or steam requirements are variable.

ECONOMICS

The R114 heat pump system is estimated to cost $761,000. The steam compression heat pump is estimated to cost $465,000 for the centrifugal compressor; $565,000 for the screw compressor. Including a $16,000 installation cost, the total capital cost would be $1,226,000 (centrifugal steam compressor) or $1,326,000 (screw steam compressor). If a stainless steel condenser is specified, the additional cost would be $176,000.

Potential savings were estimated based on process utilization of 80% and a scheduled operation of 24 hours per day, 7 days per week, 50 weeks per year. Savings of $426,000 per year were predicted, where

DATA:

Waste heat source:	White water
R114 heat pump COP:	4.5
Steam compression heat pump COP:	8.3
System output:	4.5 MW
COP:	3.1
Installed cost:	$1,226,000
Savings:	$426,000/year
Payback:	2.9 years

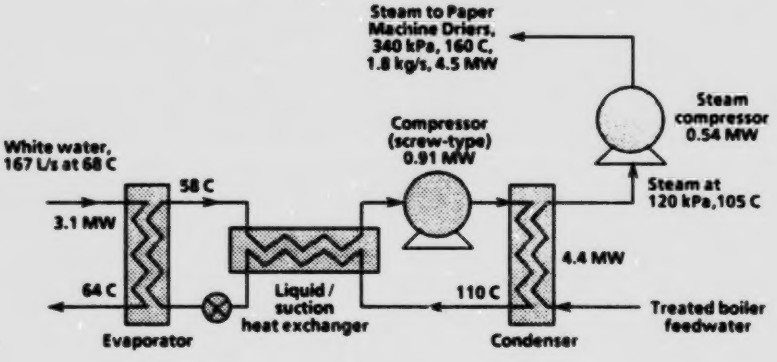

Steam to Paper Machine Driers, 340 kPa, 160 C, 1.8 kg/s, 4.5 MW
Steam compressor 0.54 MW
Compressor (screw-type) 0.91 MW
White water, 167 L/s at 68 C
58 C
3.1 MW
Steam at 120 kPa, 105 C
4.4 MW
64 C
Liquid / suction heat exchanger
110 C
Treated boiler feedwater
Evaporator
Condenser

Figure 1. Design of a Steam Generating Heat Pump, Combining an R114 Heat Pump with a Steam Compression Heat Pump.

A white water stream from a pulping machine could provide the heat source. The R114 heat pump produces low pressure steam, which is upgraded using an open cycle steam compression heat pump, to produce 340 kPa medium pressure process steam.

electricity costs 2¢/kWh and steam would otherwise be generated from oil fired boilers at a cost of $6/GJ.

A simple payback period of approximately 2.9 years was predicted for the centrifugal steam compressor option; 3.1 years if the screw compressor option was selected.

CONCLUSIONS

For installations where part load operation will be infrequent, the heat pump system incorporating a centrifugal steam compressor is the less expensive design. For installations where capacity reductions of less than 70 percent of full load will occur frequently, the system incorporating a screw steam compressor may be less expensive in the long run because of reduced steam compressor operating costs.

UPDATE

This study was critiqued and the economics of the system revised in a subsequent report summarized on p. 2-13.

CRITIQUE OF A STEAM GENERATING HEAT PUMP DESIGN

G.A. Robb Associates of Ottawa completed this critique and application study in March 1986.

PROJECT RATIONALE

In the previous project digest, the design of a steam generating heat pump was presented (p. 2-11). The design was based on waste heat available and process steam requirements at a particular pulp and paper plant. The design, however, would have broad application to a wide range of industrial plants which use large quantities of process steam.

The objectives of this study were to:

- provide a critique of the design; and
- to provide two illustrative examples of its application in pulp and paper plants.

CRITIQUE OF DESIGN

A skid mounted steam generating heat pump was designed, comprised of an R114 closed cycle heat pump for producing low pressure steam coupled with an open cycle steam compression heat pump (steam compressor) capable of upgrading the steam from low pressure (120 kPa) to medium pressure (340 kPa) for industrial process applications. The system is shown schematically in Figure 1.

On the whole, the design was found to be sound. Three suggested improvements to the design were cited.

The optional stainless steel material specification for the condenser / steam generator (in lieu of carbon steel) should be selected. Not only would there be significantly less refrigerant decomposition on the shell side as described in the original design report, there would also be significantly less corrosion on the tube side where the steam is generated. In the case of carbon steel, tube corrosion could be a serious problem unless makeup water is deaerated and an oxygen scavenger added. Additionally, rapid corrosion can occur when the unit is shut down.

The design calls for a solenoid valve to control makeup water to the generator. This implies on / off control which would presumably cause pressure fluctuations unless the makeup water is at the temperature of the steam in the receiver / steam separator. Since the makeup water will be at a lower temperature, continuous throttling control would be more appropriate.

Finally, some provision for periodic flow reversal of the white water in the evaporator for a short period

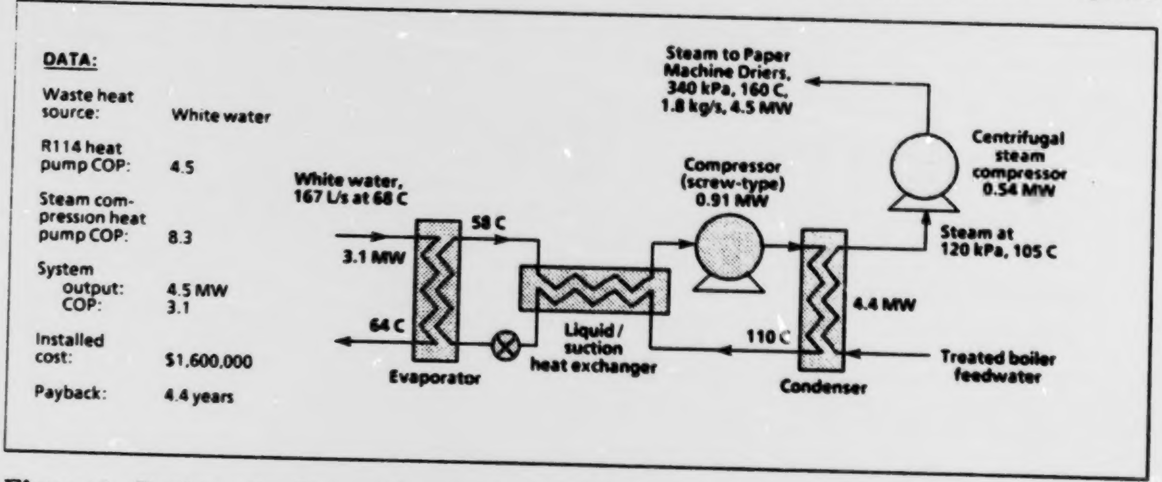

DATA:
Waste heat source: White water
R114 heat pump COP: 4.5
Steam compression heat pump COP: 8.3
System output: 4.5 MW
COP: 3.1
Installed cost: $1,600,000
Payback: 4.4 years
Steam to Paper Machine Driers, 340 kPa, 160 C, 1.8 kg/s, 4.5 MW
Centrifugal steam compressor 0.54 MW
Compressor (screw-type) 0.91 MW
White water, 167 L/s at 68 C
58 C
3.1 MW
Steam at 120 kPa, 105 C
4.4 MW
64 C
Liquid / suction heat exchanger
110 C
Treated boiler feedwater
Evaporator
Condenser

Figure 1. Design of a Steam Generating Heat Pump, Combining an R114 Heat Pump with a Steam Compression Heat Pump.

A white water stream from a pulping machine could provide the heat source. The R114 heat pump produces low pressure steam, which is upgraded using an open cycle steam compression heat pump, to produce 340 kPa medium pressure process steam.

is recommended so that the evaporator can be backwashed if required. Experience suggests that high concentrations of fibres in the white water can occur if the white water filtering system develops problems leading to tube plugging in the evaporator.

ECONOMICS

The estimated cost for the system was originally $1,226,000. This cost, however, did not include foundations, installation labour and connecting piping. An installed cost of $1,600,000 to $1,800,000 is more realistic.

With the sudden and dramatic downturn in oil prices in early 1986, the $6/GJ is no longer realistic for the cost of steam generation. The following assumptions lead to a revised economic evaluation of the heat pump system:

- $4/GJ steam cost;
- 2¢/kWh electricity cost; and
- 7% investment tax credit.

Based on an installed cost of $1,600,000, the simple payback period would be 4.4 years and the IRR would be 20.5%.

CONCLUSIONS

One means of reducing the capital cost of the system would be to eliminate the controls which allow the heat pump system to operate at variable loads. This would only be practical where constant waste stream flows and temperatures are available and where there is constant steam demand.

Another means of reducing capital costs would be to use a single-stage centrifugal steam compressor at about half the cost of a two-stage centrifugal steam compressor. This would be feasible if the R114 heat pump was to supply low pressure steam at 170 kPa instead of 120 kPa. Steam compressor capital savings would have to be balanced against higher operating costs due to lower system COP.

Finally, where there is a demand for low pressure steam, such as in a feedwater deaerator, a low-pressure steam generating R114 heat pump will suffice, thus eliminating the capital requirement for the steam compressor.

PULP AND PAPER PLANT - STEAM GENERATION

H.A.Simons Ltd. of Vancouver, B.C. performed the following feasibility study between February and September 1984.

PROJECT RATIONALE

Drying processes in the paper industry require large quantities of energy. Waste streams exist which hold potentially recoverable energy. The Industries James Maclaren Incorporé of Buckingham, Québec was selected for a feasibility study incorporating a heat pump system which produces steam. The construction of a new paper machine at the Maclaren mill afforded the opportunity to incorporate a heat pump into the paper making process.

The objectives of the study were:

- to determine the design of a mechanical compression heat pump system for producing medium temperature steam;
- to estimate the cost and benefits of the heat pump system.

DESIGN CONSIDERATIONS

Air continuously exhausted from one of the paper machines was determined to be the best source of waste heat at the Maclaren plant. Heat exchangers were found to be the most cost effective equipment to recover energy from the exhaust air stream, capable of supplying preheated makeup air for the paper drier as well as other miscellaneous heating duties. Together, these heat exchangers would reduce the exhaust air to 49 C. Thus a saturated air stream flowing at 84.7 kg/s and 49 C would be available for heat pump heat recovery after installation of heat exchangers. For instance, 4.7 MW of energy recovery can be obtained from a further temperature drop of 5 C in the air stream.

The most economic use of a heat pump system involves its continuous operation. Two major continuous streams were identified - dryer cylinder 'pocket' ventilation and dryer hood makeup air - which could use medium temperature steam produced from a heat pump system. Pocket ventilation required 110 C, 143 kPa low pressure steam and air heating required 129 C, 262 kPa steam.

SYSTEM DESIGN

The proposed heat pump system consisted of a closed cycle refrigerant compression heat pump in series with a closed cycle steam compression heat pump. This system is shown in Figure 1.

The refrigerant cycle heat pump would use R114 and would require 2 stage compression using centrifugal compressors. Steam at 93.5 C, 80 kPa would be produced at the condenser. It would be upgraded to 110 C, 143 kPa using a centrifugal steam compressor. Some of this steam would be used to heat the makeup air for the paper machine. The balance of the steam would be further upgraded to 129 C, 262 kPa using a second centrifugal steam compressor, where it would be used to provide pocket ventilation heating.

A COP of 2.9 was calculated for this system, based on a system output of 4.6 MW and an energy input of 1.6 MW.

ECONOMICS

Steam generated by the heat pump system would displace steam presently produced by electric boilers. The saving on the energy cost was estimated to be $600,000 per year. Based on a capital cost of $2,200,000, the payback period would be 3.6 years.

CONCLUSIONS

A heat pump system for producing medium grade steam at 262 kPa was predicted to have a simple payback period of 3.6 years.

UPDATE

Subsequent to the completion of this study, the heat pump system was not incorporated into the new machine because of scheduling and corporate policy reasons.

DATA:

Process:	Steam production for air heating
Waste heat source:	Newsprint machine hood exhaust
Input:	1.6 MW
Output:	4.6 MW
System COP:	2.9
Capital cost:	$2,200,000
Savings:	$600,000/year
Payback:	3.6 years

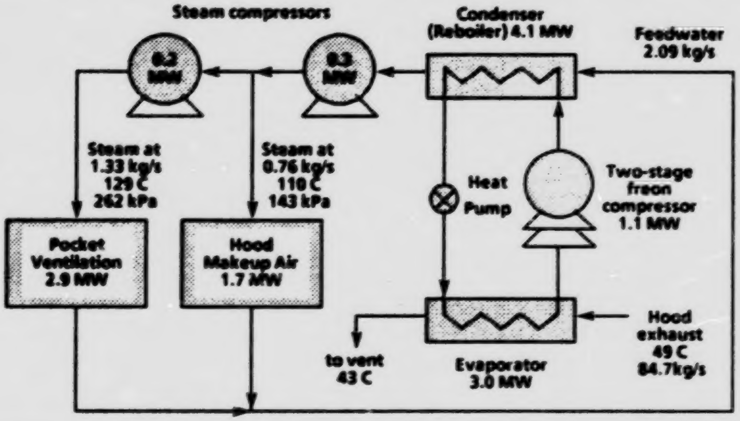

Steam compressors
0.2 MW
0.3 MW
Condenser (Reboiler) 4.1 MW
Feedwater 2.09 kg/s
Steam at 1.33 kg/s 129 C 262 kPa
Steam at 0.76 kg/s 110 C 143 kPa
Heat Pump
Two-stage freon compressor 1.1 MW
Pocket Ventilation 2.9 MW
Hood Makeup Air 1.7 MW
to vent 43 C
Evaporator 3.0 MW
Hood exhaust 49 C 84.7kg/s

Figure 1. Exhaust Heat Recovery Using Heat Pumps, at James Maclaren Industries.

Heat is recovered from the hood exhaust of a newsprint machine by means of a closed cycle refrigerant compression heat pump in series with a closed cycle steam compression heat pump. The steam generated by the system is reused in the newsprint machine.

DAIRY - BOILER FEEDWATER HEATING

The Centre de Récherche Industrielle du Québec performed this feasibility study during the period July 1981 to May 1983.

PROJECT RATIONALE

Large quantities of low temperature condensate are drained to sewer in the dairy industry. Heat pump technology provides significant potential for recovering heat from this condensate.

The objectives of this project were:

- to study the feasibility of installing a heat pump heat recovery system in a dairy;
- to provide an economic analysis of the system and establish the market potential for such systems in the Canadian dairy industry.

SYSTEM DESIGN

A detailed analysis was undertaken of the potential waste heat streams available at the Agrinove dairy in Ste. Claire, Québec, owned by the Coopérative Laitiére du Sud de Québec. The condensate from the milk evaporation process was the most suitable waste heat source. The condensate was found to be available continuously, at a flow rate of 19 L/s and temperature of 38 C in the winter, and at 39 L/s and 49 C in the summer.

The proposed heat pump system would recover heat from this waste stream and supply preheated feedwater to the dairy's steam boilers. The feedwater, from the local river, was found to range in temperature from 2 C to 24 C, depending on season.

The heat pump selected was a Templifier unit. This R114 heat pump with ancillary equipment was available 'off-the-shelf'.

The proposed system comprised a heat exchanger and a heat pump, with the boiler feedwater passing in series through each unit (see Figure 1). Because of the wide temperature differential between the incoming feedwater and the waste condensate, a heat exchanger offered an efficient and inexpensive means of recovering part of the available energy. A given quantity of energy could be recovered more cheaply with a heat exchanger connected to a heat pump than with a high-capacity heat pump alone. The heat pump would supply feedwater to the boilers at about 70 C.

ECONOMICS

A preliminary analysis of the system revealed a 610 kW combined output for the heat exchanger and the heat pump. The total annual energy saving of 6840 GJ was estimated, representing a fuel cost of $33,500 (displaced oil). The economics of the system are presented below:

Total Capital Cost:		$ 94,900
Annual Net Savings:		$ 21,000
Energy Savings	$ 33,500	
Less Expenses:		
Electricity	11,500	
Maintenance	1,000	
	$21,000	
Payback Period:		4.5 years

In 1979, Canadian evaporated milk production was 135,992,000 litres. Based on the findings of this study, 0.0225 litres of fuel oil per litre of raw milk can be saved using heat pump technology. By extension, three million litres of fuel, or $480,000(1982), can be saved annually in the Canadian dairy industry.

CONCLUSIONS AND RECOMMENDATIONS

No conclusions were made.

Based on the favourable economics of the feasibility study, it was recommended that the proposed system be installed and monitored. Monitored results should be used to verify the predicted economics.

DATA:

Process:	preheating boiler feedwater
Waste heat source:	vacuum system condensate
Operating hours:	3100/year
System output:	613 kW
Heat Pump	
Output:	352 kW
COP:	3.8
Exchanger output:	261 kW
System capital cost:	$94,900
Electricity cost:	4.1¢/kWh
Fuel cost (oil):	15.6¢/L
Savings:	$21,000/yr
Payback:	4.5 years

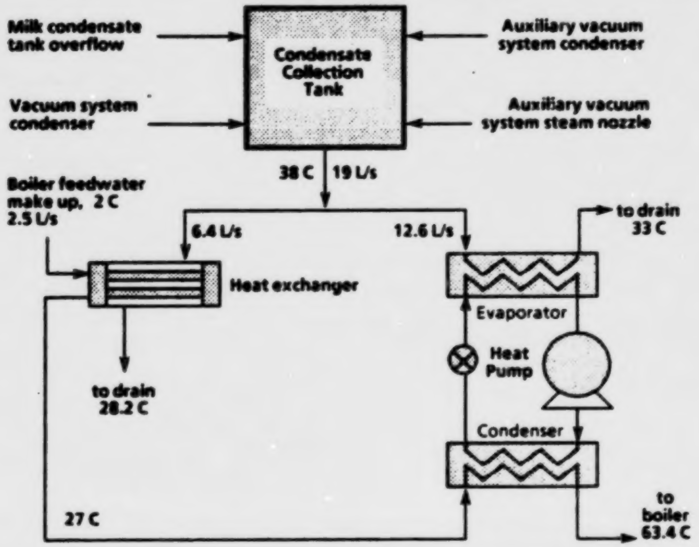

Milk condensate tank overflow
Vacuum system condenser
Condensate Collection Tank
Auxiliary vacuum system condenser
Auxiliary vacuum system steam nozzle
38 C
19 L/s
Boiler feedwater make up, 2 C 2.5 L/s
6.4 L/s
12.6 L/s
to drain 33 C
Heat exchanger
Evaporator
Heat Pump
to drain 28.2 C
Condenser
27 C
to boiler 63.4 C

Figure 1. Application of a Heat Pump in an Evaporated Milk Plant.

In the above illustration, steam from the milk evaporator condenses and is collected as hot water in the collection tank. It is then piped to the heat exchanger and the heat pump where it releases energy to the incoming fresh feedwater. The integrity and reliability of the original system are in no way affected by the modifications.

The number of operating hours (3100 hr) was established by dividing total annual milk production (28,096,000 kg/yr) by the rate of production (9072 kg/hr). The predicted annual energy reduction of 6840 GJ (worth $33,500) was calculated by multiplying the heat recovery rate (610 kW) by the annual operating hours.

DAIRY - PROCESS WATER HEATING

This feasibility study was prepared by Chalifour, Marcotte and Associés Incorporé of Pointe Claire, Québec in March 1984.

PROJECT RATIONALE

The dairy industry is one of the major energy consumers in the Canadian industrial sector. Although there are many distinct processes in a dairy, two of the most energy intensive involve moisture removal in the production of powdered milk and evaporated milk. Heat pumps have substantial potential for displacing fossil fuel generated steam in these processes.

A dairy, La Cooperative Agricole du Bas Saint-Laurent, was willing to participate in a feasibility study on heat pump applications within its facilities. A preliminary assessment of its operations indicated excellent potential for cost effective energy savings.

The objectives of this study were to:

- prepare a heat pump system design for recovery of energy from the milk evaporation process
- to determine the costs and benefits of the proposed system

SYSTEM DESIGN

After a thorough examination of process flows and temperatures at the dairy, a heat pump system was designed for recovery of waste heat from the milk evaporation process to produce process water. This process water would be used primarily for preheating air for the milk dryer, as well as providing heating or preheating for a number of other processes.

Both cooling water and condensate from the milk evaporation process is collected in a hot well before it is drained to a river. Under the proposed design, the hot well would be separated in two with a baffle. One side would collect the lower temperature cooling water while the other would collect the higher temperature condensate. Two heat pumps, designated 'low temperature heat pump' (LTHP) and 'high temperature heat pump' (HTHP) would be used to recover energy from the two halves of the hot well. Water drawn from the hot wells through the heat pump evaporators would be returned to the hot wells in order to maintain balanced flows.

The heat pump system is shown in Figure 1. The LTHP, a McQuay Templifier packaged R114 heat pump, would provide 65 C process water in a closed loop. About half of its energy supply would be used to preheat air for the milk dryer. A smaller portion would be used to boost the temperature of the source water used by the HTHP. The balance of the energy produced by the LTHP would supply a variety of processes: can sterilizer preheating, milk concentrate preheating, cleaning water production and space heating. Auxiliary steam heating would also be included in the process water loop.

The source water from hot well would be cooled by the LTHP to about 25 C. River water would be mixed with this cooled source water to provide a maximum 21 C cooling source for the milk evaporation process. After removing heat from the milk evaporation process, the water would be returned to the hot well. In this manner a constant source of warm waste water would be available to the LTHP.

The HTHP, also a McQuay Templifier heat pump, would use source water drawn from the high temperature side of the hot well. Additional heat would be added to the source water using LTHP process water. About half of the energy recovered by the HTHP would be supplied by the LTHP. The HTHP would supply 93 C process water through a closed loop. The process water would be used primarily to preheat air for the milk dryer. The heat pump would also provide heating for the can sterilizer, whole milk pasteurizer, cream pasteurizer and boiler feedwater.

Like the LTHP, source water from the hot well used by the HTHP would be returned to the hot well. in order to maintain balanced flows through the heat pump. Here it would mix with condensate from the milk evaporation process. Excess water from the HTHP evaporator would flow over the baffle to the low temperature side of the hot well.

ECONOMICS

The total capital cost is estimated to be $855,000. The annual operating savings is estimated to be $203,000, for a simple payback period of 4.2 years. The annual savings consists of $238,000 in reduced electrical cost (at 3.5¢ / kWh) less $35,000 in maintenance cost.

CONCLUSIONS AND RECOMMENDATIONS

No conclusions or recommendations were made.

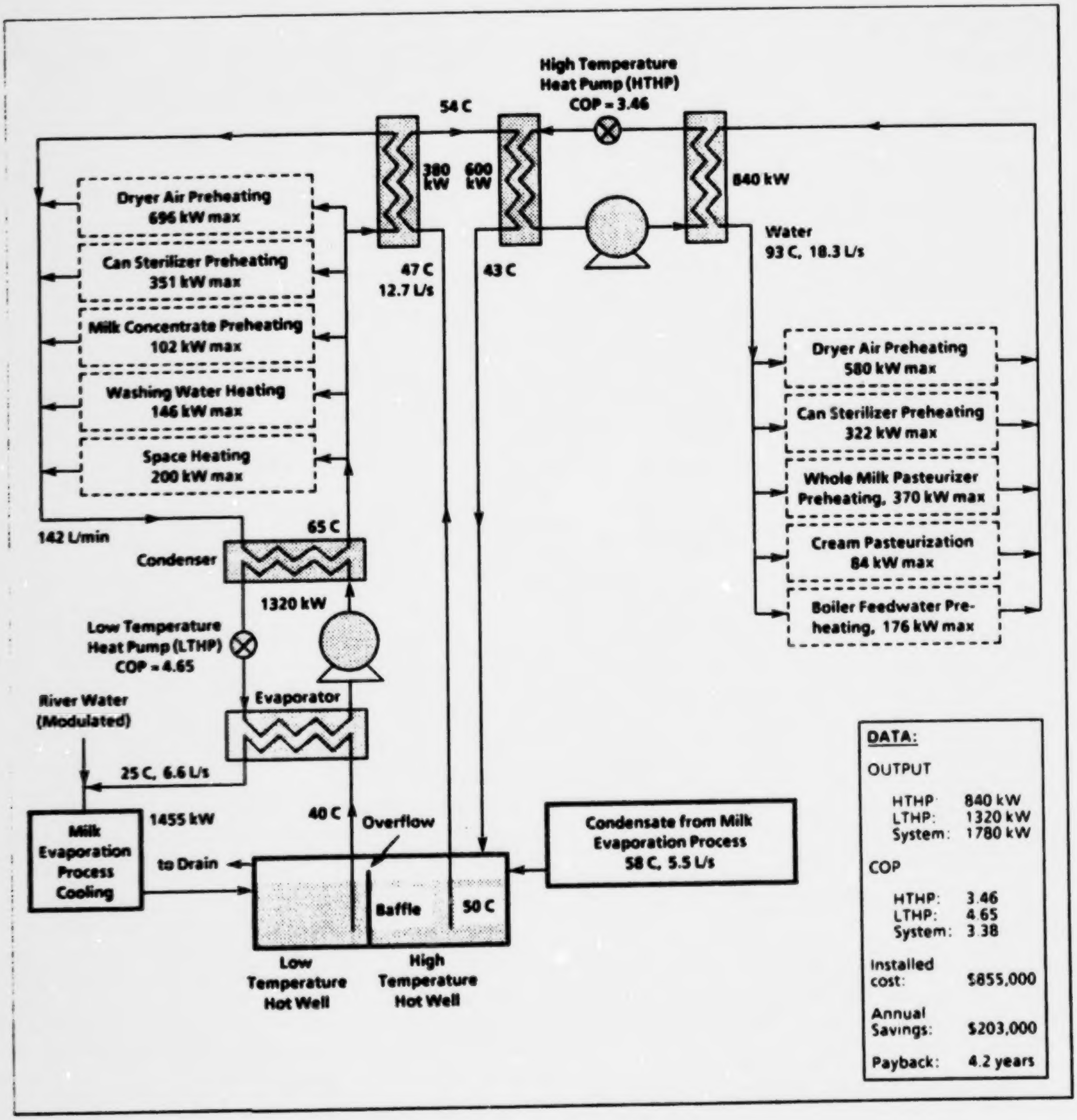

High Temperature
Heat Pump (HTHP)
COP = 3.46
54 C
380 kW
600 kW
840 kW
Water
93 C, 18.3 L/s
47 C
12.7 L/s
43 C
Dryer Air Preheating
696 kW max
Can Sterilizer Preheating
351 kW max
Milk Concentrate Preheating
102 kW max
Washing Water Heating
146 kW max
Space Heating
200 kW max
Dryer Air Preheating
580 kW max
Can Sterilizer Preheating
322 kW max
Whole Milk Pasteurizer
Preheating, 370 kW max
Cream Pasteurization
84 kW max
Boiler Feedwater Pre-
heating, 176 kW max
142 L/min
65 C
Condenser
1320 kW
Low Temperature
Heat Pump (LTHP)
COP = 4.65
Evaporator
River Water
(Modulated)
25 C, 6.6 L/s
1455 kW
Milk
Evaporation
Process
Cooling
to Drain
40 C
Overflow
Baffle
50 C
Condensate from Milk
Evaporation Process
58 C, 5.5 L/s
Low
Temperature
Hot Well
High
Temperature
Hot Well
DATA:
OUTPUT
HTHP: 840 kW
LTHP: 1320 kW
System: 1780 kW
COP
HTHP: 3.46
LTHP: 4.65
System: 3.38
Installed cost: $855,000
Annual Savings: $203,000
Payback: 4.2 years

Figure 1. Industrial Heat Pump Application In a Dairy.

PAPER DRYING

Acres Consulting Services Ltd., of Toronto, Ontario, undertook the following chemical heat pump feasibility study during the period of January to March 1984.

PROJECT RATIONALE

Paper manufacture, a major Canadian industry, uses large quantities of relatively low-grade heat for drying. Though conventional heat pumps are unsuitable for paper drying because the refrigerant fluids are unstable at the heat delivery temperatures required, Chemical Heat Pumps (CHP's) can be used effectively.

The CHP functions as an absorption drier in paper drying applications, rather than a heat pump in the accepted sense of the term. However, while the primary purpose is the transfer of mass, heat is also transferred and upgraded. Hence, the title 'Chemical Heat Pump' is considered appropriate.

The study focused on the application of the Chemical Heat Pump using sodium hydroxide as the heat transport medium. It should, however, be emphasized that many other industrial drying processes could be matched with the sodium hydroxide CHP.

The objectives of this study were to:

- design and specify CHP package and auxiliary equipment for tendering purposes, including flow sheets, site plans and equipment layout, for a paper drying process at an Ontario Paper Company plant.
- provide a commitment level capital cost estimate, schedule, cash flow projection, economic analysis, and assessment of the benefits compared to alternative heat recovery schemes.
- define the scope of required back-up R & D work and provide a cost estimate and schedule for these activities.

SYSTEM DESIGN

Flow diagrams and heat balances were prepared for the plant. Then, following the development of component design methods, plant layout configurations were established. Two sizes of plant were detailed, one representing a drying capacity equal to a full-size paper dryer, and the other representing 25 percent of this size.

Thermodynamic data was refined following a literature search and review. A mathematical expression was devised for the enthalpy of sodium hydroxide at various concentrations and temperatures. The expression was established with an accuracy of 0.2 percent or better using a computer curve fitting method. A curve fitting exercise was also carried out to determine an expression for vapour pressure as a function of temperature and concentration, with an accuracy of one-half percent or better.

The contacting tower represented the only unique component required for the sodium hydroxide CHP, and both the technical success and economic viability of the plant will be dependent upon this component. Design methods were established for a contacting tower of open type packing. A spread sheet computer program was used to carry out the lengthy heat and mass transfer calculations on a stage-by-stage basis, using thermodynamic properties established during the previous activities in the study.

A single-stage forced circulation evaporator was designed for regeneration of the diluted sodium hydroxide leaving the contacting tower.

Specification sheets were prepared for the contacting tower, the regenerator and the remaining conventional equipment of the plant.

Material properties were examined in detail and, at the temperatures and concentrations existing in the plant, pure nickel provided more-than-adequate performance. Alternatives were studied, however, in order to minimize capital costs, and the highest potential for cost reduction appears to be dual-polymer laminates. These consist of a fibreglass reinforced plastic base material, lined on the surface with a high-performance fluorinated polymer such as PTFE.

The system configuration is shown and described in Figure 1.

ECONOMIC ANALYSIS

Commitment level project costs were prepared. The capital cost of the full-size unit of 17.4 MW capacity was $2,992,000 and for the 25 percent size plant of 4.4 MW was $1,381,000.

From engineering and design through to experimental operation would require 40 months and 46 months for the 25 percent size plant and full-size plant respectively.

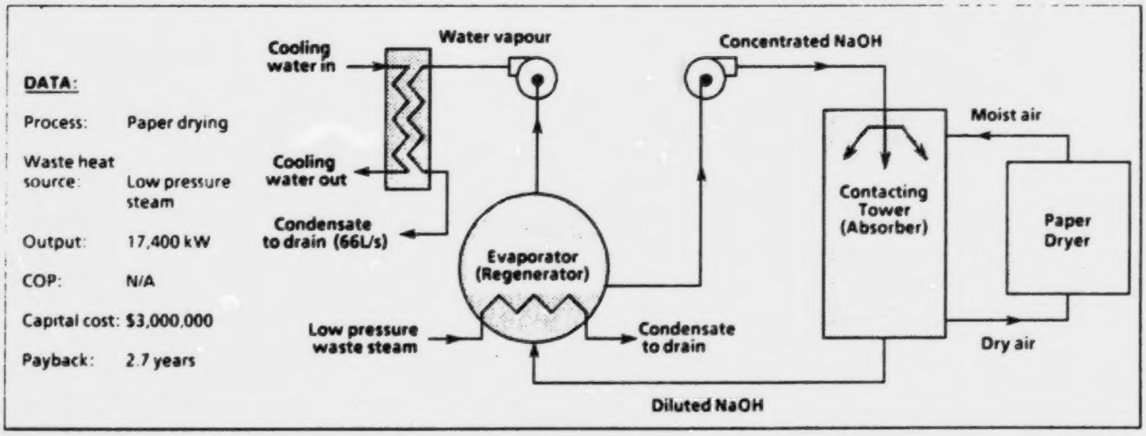
Economic analysis showed the discounted payback period to be 2.7 years for the full-scale plant and, due to the higher relative unit cost, a less favourable 6.5 years for the 25 percent plant. These figures are based on comparisons with conventional gas fired operations.

Alternative heat pump systems were compared with the CHP and heat balances were prepared for the various types. These included methanol, mechanical vapour re-compression, and thermo-compressor heat pumps. The conclusion reached was that in situations where waste steam at slightly above atmospheric pressure is available, the Chemical Heat Pump is very competitive with the other systems.

RECOMMENDATIONS

A logical next step would be to build a bench scale model of one-half percent capacity of a full-size paper dryer to verify and optimize design methods. In addition, refinement of thermodynamic data is called for and more extensive search of the materials marketplace.

In order to transfer the technology to industry, two prerequisites must be met: first a complete list of applications should be compiled; second a medium scale demonstration plant should be constructed.



DATA:

Process:	Paper drying
Waste heat source:	Low pressure steam
Output:	17,400 kW
COP:	N/A
Capital cost:	$3,000,000
Payback:	2.7 years

Figure 1. Open Cycle NaOH Absorption Chemical Heat Pump for Drying

Warm moist air from the paper drying equipment at 77 C and containing 15 percent water vapour by weight enters the top of the contacting towers. Simultaneously, sodium hydroxide solution at 80 C containing 40 percent water is sprayed into the top of the towers and is mixed with the moist air stream. A chemical reaction, exothermic in nature, occurs as the air/sodium hydroxide mixture passes down through the towers and there is a transfer of heat and water vapour between the air and the sodium hydroxide. This results in a reduction in moisture content and increase in temperature of the air, and an increase in moisture content and increase in temperature of the sodium hydroxide. The air leaves the bottom of the towers at 106 C and contains 11 percent water vapour; the sodium hydroxide is collected in a hot well at the base of the towers at 107 C and contains 42 percent water.

The warmer and dryer air returns to the paper drying equipment via a demister which removes any carried- over sodium hydroxide.

The diluted sodium hydroxide is fed to an evaporator for regeneration (i.e. removal of the moisture picked up in the contacting towers) and is subsequently recycled to the towers.

The operating pressure of the evaporator primary side is held below atmospheric pressure resulting in a low saturation temperature and consequently low sodium hydroxide outlet temperature. The regeneration medium is steam at slightly above atmospheric pressure on the evaporator secondary side, which effectively boils the excess water out of the sodium hydroxide. The water vapour driven off is condensed in a condenser cooled by service water from an available source.

GRAIN DRYING

National Research Council of Canada, Ottawa, developed the following concept study, dated July 1984.

PROJECT RATIONALE

Grain drying facilities are used to increase farm yield and revenue. Moisture content and uniformity are the most important features in determining corn or grain quality. For each one percent increment in moisture content above the 15 percent level, typically there would be a two percent reduction in the price of the grain. A non-drying farmer must harvest later which increases his field losses. Also, he must sell immediately to prevent crop mold and thus forfeits price increases of 25 percent or more which are commonly realized a few months after harvest.

Heat pump grain driers with integral heat pipe heat exchangers could be used instead of conventional fuel-fired driers and result in substantial reductions in energy costs. The heat recovery and dehumidification functions of the system would enhance the economics of the grain drying process.

The objective of this study was to outline a design approach which could be developed into a heat pump system for drying field harvested grain.

DESIGN CHARACTERISTICS

The following heat pump drier characteristics were identified:

- the heat pump should operate in a continuous cycle to obtain maximum efficiency;
- air entering the drier first cools the grain. The residual heat can be recycled and the grain immediately stored;
- grain depth should be 1 metre or less to reduce the power needed for the blower;
- the system should be adaptable to dry several grain and bean varieties to increase drier utilization;
- the maximum grain temperature should be limited to 50 C.

SYSTEM DESIGN

The proposed heat pump drier recovers both the latent heat of vaporization of the moisture removed from the grain and the sensible heat of the hot drier air. A heat pipe heat exchanger is incorporated into the design, helping reduce the required capacity of the heat pump. The system is illustrated and described in detail in Figure 1.

The heat pump compressor selection should be related to the least favourable operating conditions expected to be encountered during the drying season. Grain may arrive at 25 C with 30 percent moisture content, and ambient air may be 25 C dry bulb temperature with a 19 C dew point.

The heat pipe heat exchanger operates as a precooler of the air exiting the high temperature drying zone and not as a dehumidifier as might be supposed. It is the evaporator of the heat pump which functions as the air dehumidifier.

The air ducts should have a higher density at the beginning of the high temperature drying zone so that the grain entering the zone is brought up to high temperature as quickly as possible. As the grain gets drier in the centre and lower part of this zone, the number of air ducts can be reduced to reduce the air volume. The objective is to have all regions of this zone of the drier delivering air of high humidity to the evaporator. The path of least resistance should be down and diagonal so that concurrent flow is established and no vertical columns of grain are left without ventilation. Concurrent flow provides significantly improved drying characteristics.

ECONOMICS

The heat pump design was compared with other systems reported in the literature and showed significant energy reduction potential:

Heat pump/heat pipe design	1200 KJ/kg of water removed
Westinghouse heat pump design	1600 KJ/kg of water removed
Conventional fossil fuel drying systems	3400-9000 KJ/kg of water removed.

The proposed heat pump drier would therefore be expected to use 65 percent to 87 percent less energy than fossil fuel driers.

CONCLUSIONS

The heat pump drier with integral heat pipe heat exchanger has the potential to dry grain at a low energy input and at drying temperatures which are conducive to a high quality product. A heat pipe placed in the drying cycle will improve performance and reduce the size of the mechanical equipment.

The heat pump drier should be a new design and not a retrofit application in an existing drier design.

RECOMMENDATIONS

To determine the best design requires further investigation of the following unknowns:

- the optimum grain depth for water removal;
- the availability of sufficient or alternate electric power supplies at commercial grain drying sites;
- the number of drier locations and the market potential;
- other crops that could be processed.

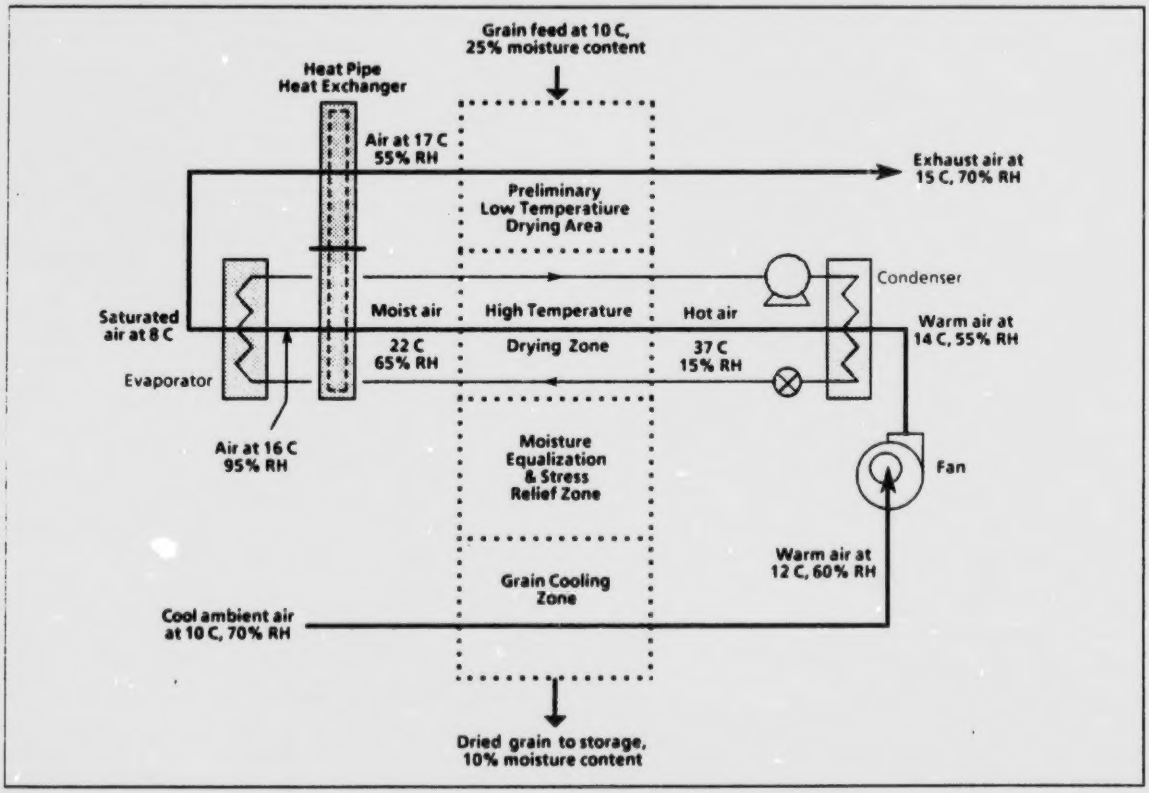

Grain feed at 10 C, 25% moisture content
Heat Pipe Heat Exchanger
Air at 17 C 55% RH
Exhaust air at 15 C, 70% RH
Preliminary Low Temperatiure Drying Area
Condenser
Saturated air at 8 C
Moist air
22 C 65% RH
High Temperature Drying Zone
Hot air
37 C 15% RH
Warm air at 14 C, 55% RH
Evaporator
Air at 16 C 95% RH
Moisture Equalization & Stress Relief Zone
Fan
Warm air at 12 C, 60% RH
Grain Cooling Zone
Cool ambient air at 10 C, 70% RH
Dried grain to storage, 10% moisture content

Figure 1. Heat Pump Application in a Grain Drying System

In the above illustration, two criss-crossing 'pathways' can be traced: one for air and one for grain.

Grain enters at the top of the diagram and moves through four zones. In the Preliminary Low Temperature Zone, high volume low temperature air provides cleaning and initial drying. In the high temperature zone, extensive moisture reduction takes place using low velocity high temperature air. The next zone provides moisture equalization and stress relief. In the last zone the grain is cooled, then transferred to storage.

Ambient air enters at the base of the grain dryer, passes through warm grain (Grain Cooling Zone), thus cooling grain and recovering heat. The warm air is blown to the heat pump condenser, where it absorbs further energy. The heated air passes through grain (High Temperature Drying Zone) where it absorbs moisture. The air then enters the heat pipe where it is cooled to saturated conditions. At the evaporator the air is further cooled, causing moisture removal through condensation. The cold dry air is then warmed by the heat pipe. This warm dry air is used for initial drying and cleaning of the grain and is then exhausted to the environment.

LUMBER DRYING

The National Research Council of Canada, in Ottawa, Ontario, developed the following concept study internally, dated March 1984. This concept was initially developed for potential application at the Industries James Maclaren Incorporé lumber kiln in Masson Québec.

PROJECT RATIONALE

Unlike conventional lumber drying which uses heated air for moisture removal, heat pumps can be used to dehumidify the air (thus increasing its capacity to remove moisture) before it is heated and sent to the drying process. Because of the lower moisture content of the air, the lumber can be dried at lower temperatures, improving the quality of the product.

The objective of this study was to develop a concept for a heat pump dehumidification and heating system for lumber kilns.

THE DRYING PROCESS

Water is present in lumber in three different locations - surface moisture, free water in the individual cells of the wood, and bound water which is in the cell walls and connecting wood fibres. This water has to be removed carefully. Cell structure in wood can be damaged and the walls of the cell can collapse if free water is removed too quickly. The bound water also has to be removed carefully so that the associated shrinking of the dryer wood does not build up stresses between the dry and the wetter wood, the cause of checking and splitting.

The drying system must be flexible in order to accommodate differing lumber loads; it must be capable of accommodating the variety of drying

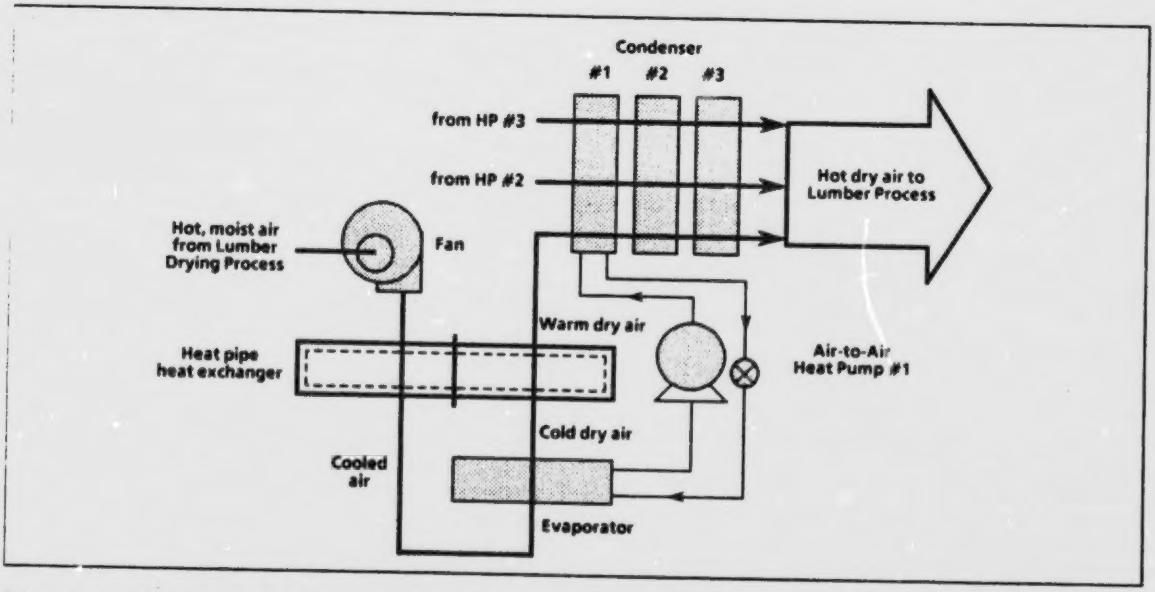


Figure 1. Heat Pump Dehumidifier for a Lumber Kiln.

Moist air from the lumber kiln is precooled at the heat pipe. It then enters the evaporator under saturated conditions, where further cooling causes the moisture to condense from the air stream. This cold dry air is warmed as it passes the heat pipe heat exchanger. The dry air becomes hot by passing through the condenser coil before being delivered to the lumber drier.

The heat pump dehumidifying unit illustrated above is identified as Heat Pump #1. This is one of three units which can operate separately or in combination, providing flexibility in the system capacity. Varying conditions of moisture content and lumber types can then be accommodated. All blowers in the system will operate continuously. This will reduce the condensing pressure and increase the COP when one or two units are operating, because the energy will be discharged into three air streams.

schedules which ensure high quality dried lumber, depending on the kind and dimensions of the wood.

SYSTEM DESIGN

The main function of the heat pump is to remove moisture from the moist air returning from the lumber stack, and that is best accomplished when the air entering the evaporator is at saturated conditions. The design selected to do this uses a heat pipe heat exchanger to precool the moist air before it enters the evaporator. The saturated air then passes through the evaporator where it is cooled below its saturation temperature, resulting in moisture condensation (dehumidification). This cold dry air is heated before it is delivered to the kiln. The system is described in detail in Figure 1.

The heat pipe heat exchanger will increase the moisture removal capabilities of the heat pump unit by as much as 50 percent compared with a heat pump of equal capacity, or allow a 25 percent reduction of system capacity for an equivalent moisture removal rate.

The proposed system would comprise three heat pump units side by side. This option would allow operation of one, two, or three systems for stepped capacity, operation of the individual compressor motors at maximum efficiency loading, and flexibility in the mixing of capacities of the individual systems. Varying conditions of the initial moisture content of the lumber and varying types of lumber can be accommodated by changing the lead compressor and the lag compressor in a combination to match the drying schedule required. The capacity of the combined operation of all compressors would match the maximum kiln drying capacity.

CONCLUSION

A heat pump lumber drier with integral heat pipe heat exchanger has the potential for drying lumber with low energy input (and hence low cost) compared with conventional driers because of combined air dehumidification and heating capabilities, and for drying at various rates conducive to high quality lumber processing.

TEXTILE DRYING

Sunwell Engineering Co., of Concord, Ontario, performed this feasibility study between August 1982 and February 1983.

PROJECT RATIONALE

A substantial amount of energy is used in industry to evaporate water in the process of drying continuous strip fabrics such as abrasive sheet,, carpet, etcetera.

The main objectives of this report were:

- to investigate the feasibility of using heat pump dehumidification on a continuous cloth drier;
- to select equipment, prepare an engineering design, and estimate the economic viability of the heat pump system.

CURRENT AND PROPOSED CLOTH DRYING SYSTEMS

Canada Sandpaper, in Plattsville, Ontario, required denim cloth for backing on one of their sandpaper products. This denim was washed, rolled, then dried in a steam-heated tenter drier. The system in place used steam at 121 C in a forced convection system to first preheat, then dry the cloth. A significant drawback was that the hot air in the drier had to be continuously exhausted to maintain the necessary relative humidity. Latent and sensible heat was lost, and pollutants from the raw cloth were exhausted into the environment.

The proposed system utilizes a heat pump. Two condensers supply the heat of evaporation; one to a conveyor belt for closed conduction preheating of the cloth, and the other to drier supply air for forced convection drying. On the evaporator side of the heat pump, moisture is removed from the drier air by condensation (see Figure 1). Both latent and sensible heat are recovered.

The plate-coil condenser would have to be custom made. It would be of stainless steel construction with a flat surface on one side and a corrugated surface on the other side. The width would be determined by the largest size of cloth in production and the length would be about 15 metres. The flat surface on top provides full contact heat transfer between the cloth and refrigerant.

ECONOMICS

Current and proposed systems economics were compared. The following is a summary of the findings based on equal water removal capabilities (213 kg/h), 4000 hours operation per year, and an average ambient temperature of -9.4 C.

THE SYSTEM IN PLACE:

Energy Use	2761 kW
Specific Energy Consumption	46,691 kJ/kg
Total Energy Cost	$154,250

HEAT PUMP DEHUMIDIFICATION SYSTEM:

Electrical Unit Consumption	245 kW
Specific Energy Consumption	656 kJ/kg
Total Energy Cost	$72,000
Total Capital Cost	$250,000

COMPARISON:

Energy Usage Reduction	83.6 percent
Energy Cost Reduction	53.0 percent
Savings Per Year	$82,250
Payback Period	3.0 years (1982 fuel prices)

CONCLUSIONS

A heat pump dehumidification system was designed with estimated energy cost savings of $82,250 per year and a payback period of 3.0 years.

UPDATE

Despite the attractive economics of this system, Canada Sandpaper decided not to invest in the heat pump system because of the capital requirement.

DATA:

Process:	Dehumidification and reheat
Waste heat source:	Warm, moist drying oven air
Operating hours:	4000/year
System COP:	3.27
Capital cost:	$250,000
Energy savings:	$82,250/year
Payback:	3.0 years

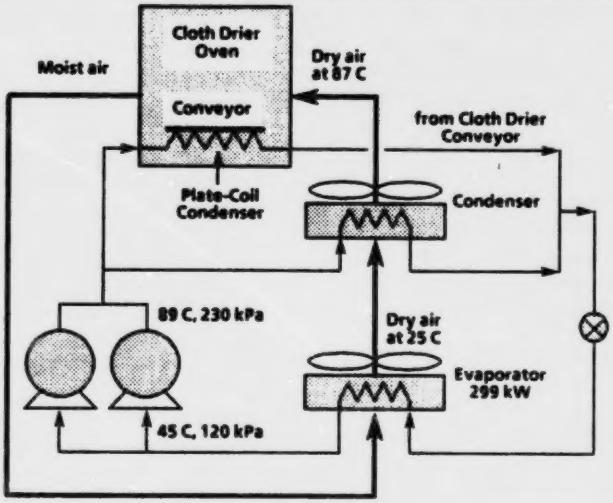

Cloth Drier Oven
Conveyor
Moist air
Dry air at 87 C
from Cloth Drier Conveyor
Plate-Coil Condenser
Condenser
89 C, 230 kPa
Dry air at 25 C
Evaporator 299 kW
45 C, 120 kPa

Figure 1. Heat Pump Dehumidification System for a Continuous Cloth Drier at Canada Sandpaper.

In the illustration above, moisture from the warm oven air is condensed at the evaporator. The two condensers, operating at 87 C, transfer heat at the preheat conveyor, and reheat the oven air stream. The refrigerant and air streams are both closed loops. No makeup air is required after these modifications, as none is exhausted.

LOBSTER MARKETING

Interprovincial Engineering, of Halifax, Nova Scotia, performed this feasibility study between August 1982 and March 1984.

PROJECT RATIONALE

Process saltwater is used in Canada's fish processing industry, and can be used as a heat source for novel heat pumps applications. The owners of Clearwater Lobster of Nova Scotia were interested in a heat pump system for salt water cooling and process water heating. The objective of this study was to evaluate the technical and economic feasibility of a heat pump system for the Clearwater facility.

SYSTEM DESIGN

Clearwater Lobster maintains a constant supply of fresh sea water to its reservoir, with an equal flow of water drained from the reservoir. The heat pump system was designed as shown in Figure 1, such that the fresh sea water is cooled to a temperature of approximately 2 C. Process water is produced at 60 C.

During the summer months when the cooling load is greatest, a heat exchanger precools the incoming sea water using the previously cooled water drained from the reservoir. The heat pump output exceeds process water requirements and excess heat is rejected through an evaporative cooler.

Selection of materials for system components must account for corrosion resistance and non-toxicity to lobsters. This narrowed the choice to titanium, plastics and glass. Because of its low cost, PVC plastic was chosen for all piping and pipe fittings. Titanium was selected for the heat exchanger and evaporator. (Glass heat exchangers cannot meet pressure safety standards and no suitable plastic heat exchangers could be found in standard production).

A plate type heat exchanger was chosen because of its high efficiency, compactness, and ease of maintenance, whereby individual plates can be removed for inspection and cleaning. Cleaning of the plate heat exchanger and the waterside of the evaporator will be required at an interval to be determined during operation.

ECONOMIC ANALYSIS

The value of the cooling of the reservoir and storage tanks on lobster production was estimated to be between $162,000 and $492,000. This was attributed to decreased cannibalism (metabolic rates are lowered in cold water), increased inventory life (allowing lucrative sales during periods of peak

DATA:

Process:	Supply process water
Waste heat source:	Sea water
Heat pump output:	614 kW
Working fluid:	R-12
COP:	3
Capital cost:	$418,000
Energy savings:	$66,000
Production savings:	$162,000
Total savings:	$228,000
Payback:	1.8 years

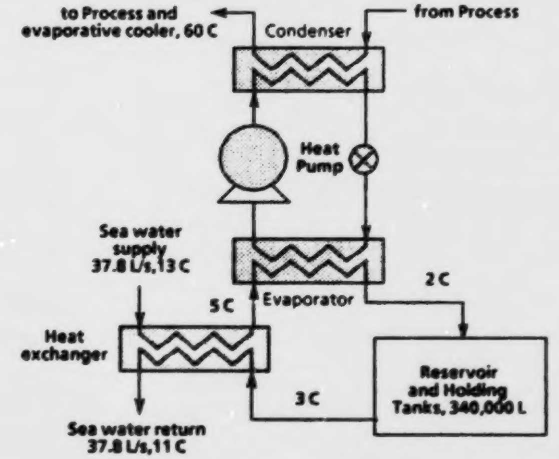

to Process and evaporative cooler, 60 C
Condenser
from Process
Heat Pump
Sea water supply 37.8 L/s,13 C
Evaporator
2 C
5 C
Heat exchanger
3 C
Reservoir and Holding Tanks, 340,000 L
Sea water return 37.8 L/s,11 C

Figure 1a. Heat Pump Refrigeration Application at Clearwater Lobsters Ltd.

During summer operation when the sea water temperature is greater than 4.8 C, incoming sea water is pre-cooled by the heat exchanger, using previously cooled water drained from the reservoir. Sample temperatures are shown. Sea water flow rate is constant year round at 37.8 L/s.

pricing) and disease control (gaffkemia, a disease fatal to lobsters, remains dormant in cold water). Process water heating costs were predicted to be reduced by $66,000. Combined annual production and energy cost benefits of a minimum $228,000 were therefore predicted. With an estimated capital cost of $418,000, the heat pump system would have a maximum 1.8 year payback period.

Economic benefits of salt water source heat pumps for other industrial applications can be inferred from this study. In this study, some of the heat produced by the heat pump is rejected to the atmosphere through an evaporative cooler. If all of the heat output (including that rejected to the atmosphere) was used for process water heating, $135,000 in energy cost savings would result. Therefore the same $418,000 heat pump in another industrial setting, where all of the heat output would be used, would have a payback period of 3.1 years.

Many industries and commercial establishments in Canada currently use or have access to large volumes of sea water on a continuous basis. In the fish processing industry it is estimated that an average plant has the capacity to pump 2.1 billion litres of sea water per year. The heat available from the total sea water flow in all the fish plants in Canada would replace the equivalent of 555 million litres of oil per year or $160 million (1983). Since most of the fish plants are located in the Atlantic Provinces this will directly reduce Canada's reliance on overseas oil imports.

CONCLUSIONS

At Clearwater Lobster, a two year payback for a heat pump system is predicted based increased lobster production and reduced energy costs. For heating applications in other industrial settings a salt water source heat pump is predicted to have approximately a three years payback period.

UPDATE

The system was installed at Clearwater Lobster and its performance is currently being monitored. A report detailing its performance and verifying its economics has been commissioned by NRCC and is expected to be completed in spring 1986.

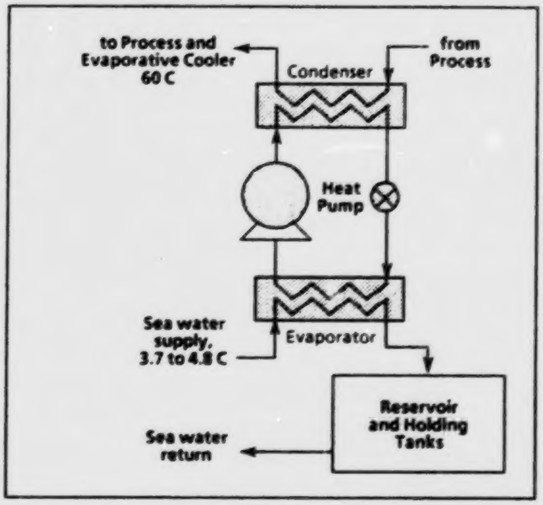

to Process and Evaporative Cooler 60 C
from Process
Condenser
Heat Pump
Sea water supply, 3.7 to 4.8 C
Evaporator
Reservoir and Holding Tanks
Sea water return

Figure 1b. Heat Pump Refrigeration Application at Clearwater Lobsters

During parts of the spring and fall seasons, precooling of the incoming sea water is not required when between 3.7 C and 4.8 C. The heat exchanger is bypassed, and the heat pump handles the entire cooling load.

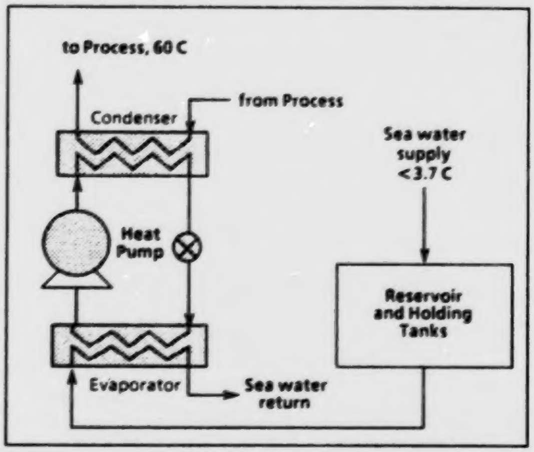


Figure 1c. Heat Pump Free Cooling Application at Clearwater Lobsters

During the winter, when the ocean temperature is less than 3.7 C, the incoming sea water bypasses both the heat exchanger and the heat pump since there is no cooling load. In order to supply the plants' process water requirements, the sea water drained from the reservoir is rerouted through the heat pump, and cooled before finally being returned to the ocean.

PHOTO PROCESSING - SERVICE WATER HEATING

James F. Hickling Ltd., of Ottawa, Ontario performed the following feasibility study between April and September 1984.

PROJECT RATIONALE

Film and paper processing machines employed in the photo finishing industry use large quantities of service hot water. After being heated and used as processor wash water, the water is drained to sewer. Recovery of energy from such wastewater to preheat incoming processor wash water is possible through heat pump technology.

The objectives of this project were:

- to present a profile of the Canadian photo finishing sector, by qualifying and quantifying energy end use data for a typical photo finishing plant and the entire service sector;
- to prepare a statement of economic viability, capital cost, benefits and payback for a wastewater heat recovery system utilizing a heat pump.

INDUSTRY SURVEY

A survey of 75 photo finishing labs and photo processing equipment suppliers was completed.

Natural gas was found to be the most popular fuel among those labs surveyed; 85% used natural gas, 10% used electricity, 5% did not respond in this area. Cost figures were unavailable, however, as lab operators had not isolated the cost of supplying hot water to their photo processing machines from total energy costs.

The photo processing industry was found to be divided as follows: conventional labs, which operate many film and paper processors; and 'mini labs', which usually have only one film and paper processor. Mini labs are commonly found in high traffic areas such as malls or department stores.

There was a significant interest in wastewater heat recovery within the industry. However, none of the mini labs and only 23% of the conventional labs employed waste energy recovery in their processes. In conventional laboratories, plate type heat exchangers were found to be not only commercially available but well suited to this application.

PLANNING AND DESIGN

It was determined that 35 percent of photo finishing labs were located in Metropolitan Toronto. A mini lab located in downtown Toronto was accordingly selected as representative of the industry.

The principal reason for selecting this site was to attain high visibility for the heat pump technology; a significant segment of the industry is located in Metropolitan Toronto and competition between operators is high.

This site also possessed all the attributes necessary to the successful installation of a wastewater heat recovery system. Access to and control of equipment and supply systems, separate metering, and natural gas water heating were present at the test site.

Preliminary engineering design involved concept development, load analysis, component selection, system integration and selection of suitable system controls. Process monitoring was also undertaken, including: DHW delivery temperature to the processors; wastewater temperature and flow rates; and city water temperature and flow rates.

Hot water supply and wastewater drainage occur continuously in photo processing operations (rather than being transient, as in the laundry and dry cleaning service sector). Storage tanks and control valving for optimizing energy recovery from the wastewater were not required. The proposed system configuration is shown in Figure 1.

Analysis of the test data allowed the heat content of the wastewater to be determined. In order to complete the test site evaluation, an energy audit was carried out. Included were building physical and operational data, utility billing information and equipment data.

ECONOMIC ANALYSIS

It was determined that the installation of heat pump wastewater heat recovery systems within the photo finishing sector is not currently economically viable. This is due to the high electricity demand charges in Ontario generally, and Toronto specifically, which results in higher operating costs compared to natural gas.

Including the electrical demand charges of using the heat pump, the utility costs are 7¢ to 9¢/kWh, compared to 1.8¢/kWh for natural gas. Discounted payback periods in the order of six to ten years were calculated using discounted analysis techniques.

CONCLUSIONS

The project established a profile of the Canadian photo processing industry, and its' positive attitude toward energy conservation issues.

Heat pump waste heat recovery systems applied to the photo finishing sector were generally found not to be feasible, based on long payback periods.

Based on the industry survey, heat exchangers are currently a preferred alternative for heat recovery.

DATA:

Process:	service water heating
Waste heat source:	film processor wash water
Heat pump output:	23 kW
COP:	3.5
Installed cost:	$4600
Payback:	6-10 years

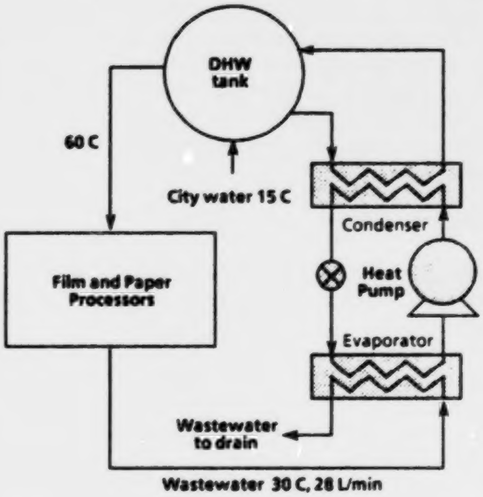

DHW tank
60 C
City water 15 C
Condenser
Heat Pump
Evaporator
Film and Paper Processors
Wastewater to drain
Wastewater 30 C, 28 L/min

Figure 1. Proposed Installation of a Heat Pump Wastewater Heat Recovery System to a Photo Finishing Operation.

Warm wash water from the processors is piped to the evaporator of a Fedders heat pump. Heat is removed from the wastewater at the evaporator. The heat pump upgrades this reclaimed heat and, at the condenser, transfers it to the fresh service water. Water at 60 C is supplied to the film and paper processors to be used as wash water.

RESTAURANT - SERVICE WATER HEATING

E.J.Faraci and Associates, of Winnipeg, Manitoba, performed the following study in two parts. The following feasibility study (phase 'A') was performed between September 1982 and March 1983. The demonstration project (phase 'B') was carried out between September 1983 and November 1984 and is summarized later in the review (p. 2-43).

PROJECT RATIONALE

According to Statistics Canada data, Canadian restaurants consumed 54 PJ of energy (1981) at a cost between $268 and $536 million. A further study, by the University of Manitoba, indicated that 10 percent of that is required for hot water heating. It was expected that this could be reduced by 65% with heat pump technology.

The objectives of this project were:

- to identify the optimum size and performance of a heat pump heat recovery system installation for a restaurant;
- to select equipment and prepare an engineering design, statement of economic viability, benefits and payback.

SYSTEM CAPACITY

The feasibility study was based on a Keg Restaurant located on Gary Street in Winnipeg. It was a full service, licenced establishment, with a seating capacity of 160 in the dining area, and 75 in the lounge.

By analyzing the utility bills, it was estimated that hot water consumption averaged 10,000 L/day. Assuming a 55 C temperature increase (10 C mains to 65 C hot water tank), the annual hot water heating requirement was estimated at 800 GJ. This represented $5600 or approximately 24% of the total energy bill (based on 1982 natural gas prices).

Hourly hot water consumption data was recorded during a two week monitoring period. Based on an analysis of the consumption data, system capacity limitations (existing hot water storage tank and maximum 50 C condenser discharge temperature) and the necessity that the heat pump operate continuously during business hours to maximize benefits, a 20 kW system capacity was determined. The capacity was later lowered to 12 kW based on two concerns. A dishwasher provides the most common hot water load in a restaurant, and if the goal is to develop a packaged, commercially available recovery system, it should be sized to the dishwater load. Hot water consumption could be reduced approximately 20 percent through implementation of other energy conservation measures.

SYSTEM DESIGN

Of immediate concern was the fact that refrigerant temperatures and pressures in the evaporator cannot be allowed to exceed the mean refrigerant temperature and pressure in the condenser. With wastewater fluctuating between 15 C and 70 C and preheat water between 10 C and 60 C, this condition would be encountered. To avoid this problem, a counterflow wastewater-to-domestic preheat water heat exchanger was installed in series with the evaporator (see Figure 1). This heat exchanger, sized to operate with a thermal effectiveness of 50% or more, would guarantee that the wastewater temperature would always be equal or lower than that of the preheat water. It would also increase energy savings by transferring a large portion of the waste heat with a very low energy input.

A Fedders packaged heat pump was found to be well suited to this wastewater heat recovery system. With its rotary compressor based design, using R12 instead of R22, together with a double wall coaxial counterflow heat exchanger (large temperature differences between water and refrigerant) evaporator and condenser water temperatures could be equal without operating problems.

ECONOMICS

Reliability and maintenance factors presented the greatest 'unknowns' associated with the proposed heat pump application. Although a ten year life expectancy for heat pump compressors would normally be accepted, one operating sixteen hours a day under a wide variety of conditions might not achieve this standard. Also, research indicated the positive emulsifying effect of dishwater detergent would prevent grease fouling of the evaporator coil. Only the operation of a prototype system, however, would prove this. Operations would also determine the required frequency of tank cleaning and of back washing. Energy recovery potential could hopefully offset a reasonable increase in maintenance and equipment replacements (e.g. weekly cleaning, and replacement of the compressor every five years)

DATA:

Process:	Service water heating
Waste heat source:	Restaurant wastewater
Refrigerant:	R12
Heat pump output:	12.8 kW
COP:	4.5
Installed cost:	$14,700
Energy savings:	$ 3,520
Payback:	4.2 years

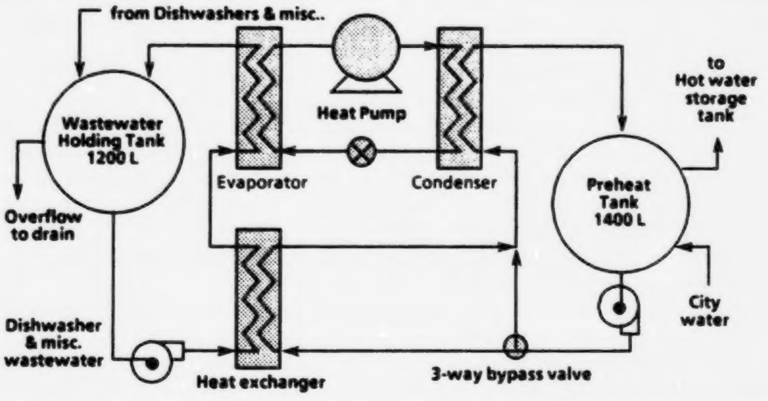

from Dishwashers & misc..
Heat Pump
to Hot water storage tank
Wastewater Holding Tank 1200 L
Evaporator
Condenser
Preheat Tank 1400 L
Overflow to drain
City water
Dishwasher & misc. wastewater
Heat exchanger
3-way bypass valve

Figure 1. Proposed Restaurant Wastewater Heat Recovery System Utilizing a Heat Pump.

When restaurant operations commence in the morning and a DHW heating load is created, the heat pump is activated to provide DHW preheat. The heat exchanger is bypassed, otherwise the lower temperature wastewater would rob heat from the preheat tank.

After several hours of operation, the temperature in the preheat tank would begin to drop below the temperature in the wastewater holding tank. Temperature sensors would signal the automatic diversion of water from the preheat tank through the heat exchanger. The heat exchanger transfers a large portion of the heat from the wastewater with very low energy input.

Engineered systems would not be cost effective for restaurant installations. Off-the-shelf packages would be required to meet an acceptable payback period of two or three years. Estimated installed cost of such a package in a retrofit situation was expected to be $14,700 ($4000 for the packaged unit and $10,700 for installation) and have a payback of 4.2 years. The economics are detailed in Figure 1.

The installed cost of a packaged heat pump system in a new restaurant would be significantly lower because the additional costs of retrofit would be eliminated. It was expected that installed cost would be under $8000 in new installations, which would reduce the payback to 2.3 years.

POTENTIAL FOR PROPOSED TECHNOLOGY

The 75 Keg Restaurants in Canada could recover 30,000 GJ of energy which represents an energy cost saving of $260,000 per year.

Figures for the Canadian restaurant industry were predicted as follows:

Energy consumed for hot water heating	5.4 PJ
Cost of energy consumed for hot water heating	$27 to $54 million
Predicted savings (65% of $27 to $54 million)	$17.5 to $35 million

CONCLUSIONS AND RECOMMENDATIONS

It was concluded that heat pump heat recovery systems for restaurants would be cost effective, with payback periods of 4.2 years for retrofit and 2.3 years for new restaurants.

The widespread application of the proposed technology in restaurants would require the development and marketing of a proven heat recovery package.

It was recommended that this engineering design phase be followed with a demonstration phase to verify predicted costs (including maintenance costs) and benefits.

EDIBLE OILS INDUSTRY - BOILER FEEDWATER HEATING

Monarch Fine Foods Co. Ltd. of Rexdale, Ontario undertook this internal demonstration study between October 1981 and April 1983.

PROJECT RATIONALE

An opportunity was afforded for performance evaluation of a newly installed industrial heat pump system at Monarch Fine Foods Co. Ltd., Rexdale, Ontario. Data collected through monitoring and recording instruments would be used as a basis for evaluation.

The objectives of this study were:

- to report on the installation of an industrial heat pump and associated equipment;
- to review the economic viability of the heat pump system with respect to predicted and proven energy cost benefits.

PROJECT OVERVIEW

Monarch Fine Foods began operating a heat pump waste heat recovery system in March of 1982, using process cooling water as a heat source for preheating boiler feedwater.

At the onset of this project, sophisticated electronic monitoring equipment, provided by Ontario Hydro and Monarch Fine Foods, was incorporated into the system. The equipment recorded operating characteristics of the heat pump and provided a means to evaluate process energy savings.

The monitoring equipment recorded the following data: source water and boiler feedwater flow rates and temperatures; refrigerant temperatures and pressures in the heat pump circuit; and system electrical consumption.

HEAT SOURCE

The heat source was cooling water drawn from deodorizers, hydrogenation convertors and continuous and batch refinery operations.

Significant features of the source water flow can be summarized as follows:

- the source water flow rate changed in a step-wise fashion due to the cyclical operation of one deodorizer;
- source water temperatures fluctuated, for a variety of reasons. Production variations in batch processes that return hot water directly to the cooling tower and cyclical batch cooling of a deodorizer were the most common causes of the temperature fluctuations.

Flow rates of the boiler makeup water were generally between 6.3 and 12.6 L/s. Temperatures varied seasonally from a low of 1.1 C to a high of 15.5 C.

OPERATIONAL EXPERIENCE

The monitoring process was inhibited because significant problems were encountered with the heat pump unit itself or its ancillary equipment. Initial 'tuning' of the unit proved impossible because of severe heat source and heat sink flow fluctuations. Though the heat pump functioned, the rate of heat recovery was below that expected.

Later, the analyzer for detecting heat pump lubricating oil in the boiler feedwater stream failed. Repairs were lengthy and during the delay refrigerant migrated into the lubricating oil. This created problems in maintaining oil pressures at startup. The oil was replaced and the system operated well.

The system was restarted after a scheduled shutdown. The oil detection analyzer in the boiler feedwater stream tripped the automatic shutdown of the heat pump, however. Iron deposits were found to be the cause of the analyzer failure, rather than the detection of oil. In order to avoid a recurrence of the problem, new piping and corrosive resistant sealants on the condenser heads and tubesheets were installed.

Shortly after the second restart, the compressor motor inexplicably failed. During the replacement period, refrigerant again migrated into the lubricating oil, requiring a fresh recharge of oil before restarting the system. In order to avoid subsequent problems with refrigerant contaminated oil after prolonged system shutdown, an oil heater was added.

Then, mechanical problems with the cooling towers resulted in the source water temperature falling too low for the heat pump to operate. The system was finally restarted a month later and it operated well for another month, before a thermistor failed. This was followed closely by the failure of the anti-cycle timer and repairs took until the end of the study period.

Ignoring scheduled interruptions, the heat pump operated only 48 percent of available hours.

ECONOMICS

The heat pump monitoring provided the data to calculate energy cost savings of $ 93,000 per year. With a capital cost of $224,000, the payback period was 2.4 years.

CONCLUSIONS

It was concluded that the heat recovery system proved worthy of the capital invested.

The total energy value added to the boiler feedwater system amounted to $98,653. It is noteworthy that the heat pump accounted for a net savings of only $9,303, with the heat exchanger supplying the remainder. The average rate of heat recovery by the system was only 322 kW, with a system COP of 4.85 (compared to predicted figures of 586 kW and a COP of 5.2). The simple payback period was 2.4 years. On an individual basis the return on the heat pump would be 15.1 years and the heat exchanger 1.1 years.

UPDATE

The heat pump was permanently valved out of the system in August 1985, one year after this study was completed, due to maintenance problems and expenses.

DATA:

Process:	Boiler feedwater preheating
Waste heat source:	Plant processes
Load factor:	48%
Output	
avg. system:	322 kW
heat pump:	32 kW
System COP:	4.85
Installed cost:	$224,000
Electricity cost:	3.3¢/kWh
Displaced energy cost:	1.9¢/kWh
Energy savings	
system:	$99,000
heat pump:	$9000
Payback:	2.4 years

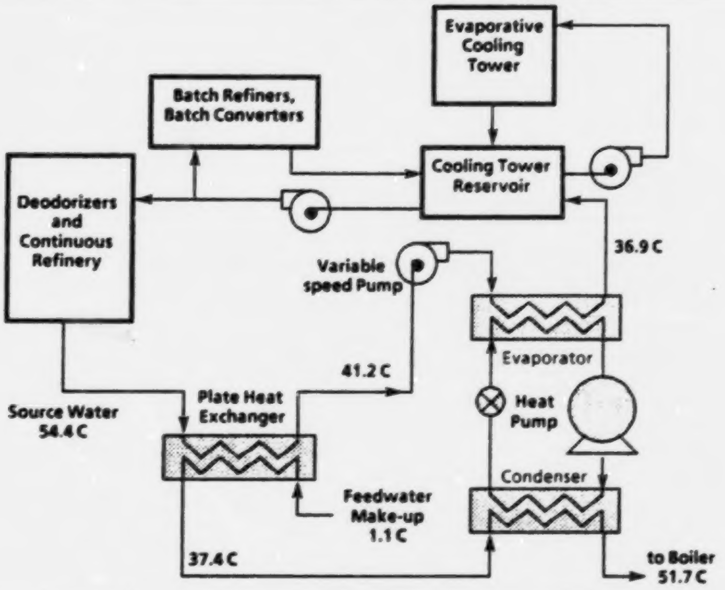

Evaporative Cooling Tower
Batch Refiners, Batch Converters
Cooling Tower Reservoir
Deodorizers and Continuous Refinery
Variable speed Pump
36.9 C
Evaporator
41.2 C
Heat Pump
Plate Heat Exchanger
Source Water 54.4 C
Condenser
Feedwater Make-up 1.1 C
37.4 C
to Boiler 51.7 C

Figure 1. Heat Pump Application at Monarch Fine Foods Limited

In the Monarch Fine Foods heat recovery system illustrated above, heated source water leaves the plant processes and is piped to the inlet of the heat exchanger. As the source water exits the heat exchanger, a variable speed pump delivers a portion or all of this stream to the evaporator of a Templifier heat pump. On the condenser side of the heat pump the full flow of boiler feedwater is allowed to pass.

The source water now returns to the cooling tower reservoir for reuse and the heated boiler feedwater is brought to the boilers for conversion to steam.

Operation of this heat pump was terminated in August 1985 because repeated equipment failures entailed unmanageable expenses.

SMELTING INDUSTRY - BOILER FEEDWATER HEATING

Brunswick Mining and Smelting Corp., of Belledune, N.B., undertook this internal demonstration project between December 1983 and September 1984.

PROJECT RATIONALE

A heat pump system, designed in-house, was installed and operated at Brunswick Mining and Smelting during the above contract period.

The objective of this project was to demonstrate the capability of a heat pump to provide process hot water by recovering a large portion of the energy which is otherwise dissipated to the atmosphere through a cooling tower.

SYSTEM DESIGN

The heat pump system utilizing a Templifier packaged heat pump was designed to preheat city mains water for process application (eg. boiler make-up water and service hot water for propane vaporization). In this case the water was drawn through a heat exchanger and the condensers of two heat pumps. Source heat was a second water stream originating in the cooling jackets of the smelting furnaces. Though linked in series, the heat exchanger and heat pump units could be isolated - a feature that was incorporated through complex valve placement and piping. The system is described in detail in Figure 1.

COMMISSIONING AND OPERATION

Installation was completed without any significant problems. Design problems, however, seriously impaired operations. Interruptions continued throughout the project due to the inability of the system to cope with flow variations in make-up water and temperature variations in the source heat stream.

The more serious problem was the flow variations. To combat instantaneous changes in flow a recirculation loop was installed. This kept a nearly constant flow of make-up water available.

Source water temperature variations caused high pressure cut-out in the heat pumps. The situation was helped by replacing the compressor valve with one which had a higher pressure cut-out sensor. However, problems associated with demand flows and source temperature fluctuations continued.

Inadvertent lowering of heat pumps' refrigerant content led to the discovery that, at lower capacity, the heat pumps maintained their efficiency. Operating at 60% capacity the pumps drew 60% power. Lower demands on the system during the fall and spring months could be addressed by this - releasing refrigerant from one pump at a time until an optimum level is reached.

Equipment problems plagued the system. A major problem was a compressor failure. It was replaced under warranty, but delivery was delayed for four months. Other failed components included circulation pump motor couplings, pump bearings and solenoid valves.

Despite an intricate monitoring system, data analysis was limited due to interruptions in the operation of the heat pump system and malfunctions in the data logger. A computer file containing 33 hours of data, recorded with all components on-line, was analyzed. Graphical analysis was prepared using this available data and used to establish the performance of the system. A COP of 10.5 was determined for the system (including the heat exchanger). The low temperature and high temperature heat pumps had COP's of 5.4 and 4.5 respectively.

ECONOMICS

The capital cost of the project was $222,746. The costs would have been higher had outside expertise been employed for the design, installation and monitoring of the system.

The system was designed to operate 8 months of the year, displacing $47,300 of oil heating costs. Estimated operating costs for the heat pump system were not given in the report. However, if one assumed an approximate equivalence between oil and electrical heating costs and a system COP of 10.5, operating cost can be estimated at $4500, with a net energy cost savings $42,800. This estimate would result in a 5.3 year payback period for the system.

CONCLUSIONS

The system had many commissioning problems, making economic analysis difficult. No payback period was reported. However, considering data presented in the report, a payback period of just over 5 years can be calculated.

DATA:

Process:	Boiler make up water
Waste heat source:	Blast furnace cooling jacket
Output:	424 kW
COP No.1:	5.4
COP No.2:	4.5
System COP:	10.5
Capital cost:	$223,000
Savings:	$42,000/year
Payback:	5.3 years

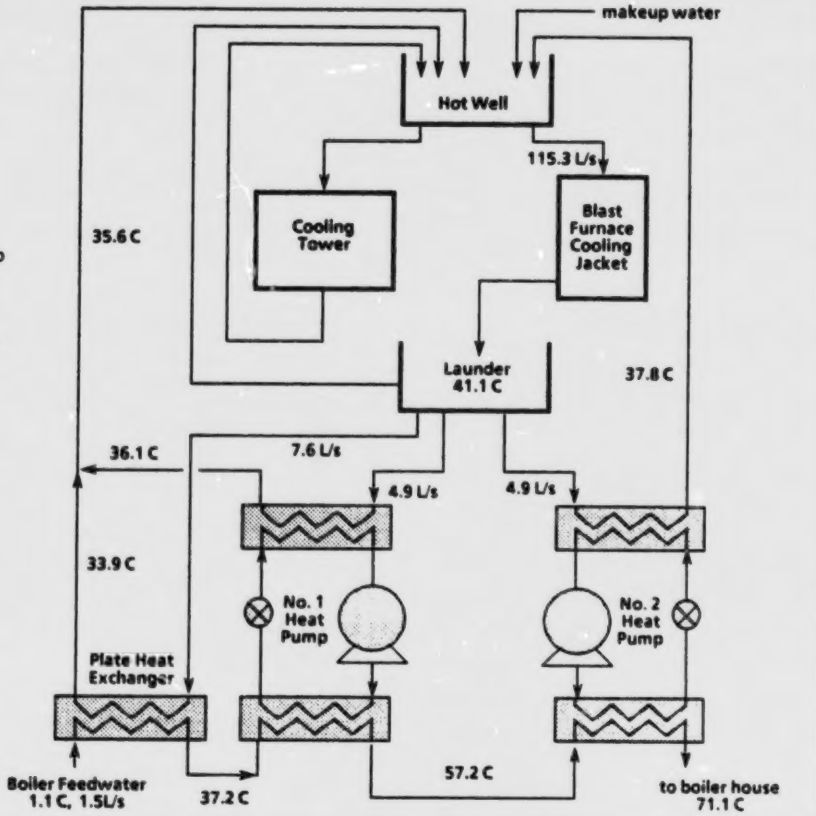

makeup water
Hot Well
115.3 L/s
Cooling Tower
Blast Furnace Cooling Jacket
35.6 C
Launder 41.1 C
37.8 C
36.1 C
7.6 L/s
4.9 L/s
4.9 L/s
33.9 C
No. 1 Heat Pump
No. 2 Heat Pump
Plate Heat Exchanger
57.2 C
Boiler Feedwater 1.1 C, 1.5L/s
37.2 C
to boiler house 71.1 C

Figure 1. Heat Pump Application at Brunswick Mining and Smelting

A flow diagram for the waste heat recovery system is shown above. Cooling water is pumped through the blast furnace from a central hot well. The cooling water leaving the blast furnace is collected in a launder. Part of this warm water is pumped in parallel to a plate heat exchanger and the evaporators of two heat pumps before it is returned to the hot well.

Boiler feedwater make-up passes in series through the plate heat exchanger and the condensers of the two heat pumps. The hot feedwater is transferred to the boiler house to be converted to steam or used as process water. The cooling tower is used to balance the hot well cooling load.

A stainless steel counterflow plate type heat exchanger is used. The two heat pumps are McQuay Templifier models TPB-025A and TPB-025B. Both heat pumps have two reciprocating compressors. The low temperature unit is charged with refrigerant 500, an azeotrope of dichlorodifluoromethane and difluorethane, and the high temperature unit with refrigerant 12.

The temperatures and flow rates shown in the above schematic are a set of typical values monitored at the plant. There can be significant variations in these values on a day-to-day basis.

PAPER PLANT - BOILER FEEDWATER HEATING

This demonstration project was undertaken by CIP Incorporated and documented in a report dated March 1986.

PROJECT RATIONALE

The objective of this study was to document the performance of a heat pump heat recovery system in a paper plant.

SYSTEM DESIGN

The heat pump system consisted of two heat exchangers and a heat pump. Make-up water for the boiler plant is progressively heated as it passes through the heat exchangers and the heat pump condenser. Water, in excess of the requirements of the boiler plant, is used in a paper machine. Heat is recovered from "white water". White water, removed from the pulping process, is contaminated with small concentrations of paper fibre. A schematic of the heat pump system is shown in Figure 1.

The heat pump was a McQuay 'Templifier' R114 packaged heat pump. The centrifugal compressor had a capacity of 425 kW.

SYSTEM MONITORING

The system was installed in April 1984.

Data acquisition hardware was incorporated in the system. Temperatures, pressures and flow rates of the fresh water and the white water were monitored. Also included in the monitoring were the heat pump power and energy consumptions. An integrated computer was used to manipulate the data so that parameters such as COP would be calculated automatically.

Although installed to coincide with system start-up in April 1984, the data acquisition system did not function properly until the following December. A vast array of problems were encountered; the primary problem, however, was that technicians were unavailable, causing delays in diagnosing and correcting problems.

SYSTEM PERFORMANCE

The heat exchangers were operated without incident between April 1984 and May 1985, when they were opened for inspection. The white water side of the heat exchangers showed no sign of fouling. The fresh water side was noticeably fouled, however. Because the heat exchangers were of spiral design, only limited mechanical cleaning was possible. Nonetheless, a 16% increase in heat transfer and a 9% increase in the fresh water flow rate was determined, based on monitored data before and after the cleaning. In July 1985, a two-stage chemical treatment was undertaken, after which a 50% improvement in energy recovery was noted.

The heat pump was operated in series with the heat exchangers, and could be isolated from the heat exchangers for service and maintenance. The heat pump operated with difficulties for 10 months, at which time McQuay representatives finally were able to make recommendations, which resolved the

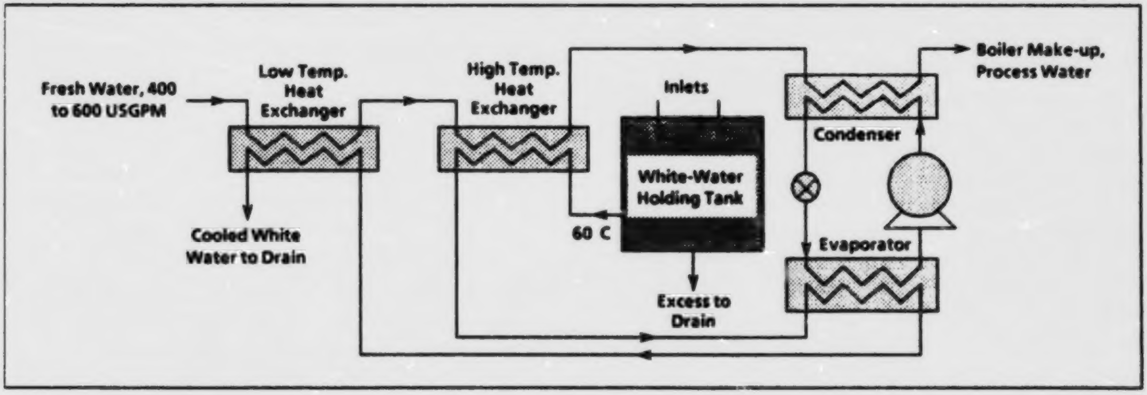

Fresh Water, 400 to 600 USGPM
Low Temp. Heat Exchanger
High Temp. Heat Exchanger
Inlets
Boiler Make-up, Process Water
Condenser
White-Water Holding Tank
60 C
Cooled White Water to Drain
Evaporator
Excess to Drain

Figure 1. Schematic of a Heat Pump Heat Recovery System used to provide boiler feedwater and paper machine process water using white water as a waste heat source.

problems. The problems centred around the oil cooling and oil heating systems, which made the start-up of the unit difficult at times and impossible at others. New heating elements were added to the oil reservoir. Oil temperature controls were adjusted to keep the oil temperature within 24C of the evaporator white water and thus prevent the refrigerant from condensing in the oil and diluting lubrication. Also controls were adjusted to increase the temperature range of the oil, from a 43C to 66C range to a 32C to 71C range; oil could then migrate to the compressor more quickly and the period required for reheating the oil was reduced from 2-6 hours to 20-30 minutes. It was speculated that these problems had contributed to the compressor failure experienced after 2 months of operation and the many oil pump failures during the 10 month period.

Aside from mechanical problems, large variations in temperatures and flow rates of both the treated water and the whitewater lead to unbalanced conditions in the heat pump and attendant stoppage. In the case of the treated water, boiler feedwater was not required on a steady basis. A secondary use for the treated water was found. Water in excess to boiler feedwater requirements was sent to a reservoir for use in one of the paper machines. A recirculation loop was installed on the whitewater waste stream to direct it back to the holding tank as required.

After the above modifications were made, the heat pump functioned well, and commissioning of the system was thought to be complete. After a month of operation, however, refrigerant pressure began to gradually drop. The system was recharged, but the pressure continued to fall. As no major external leaks were found, the condenser and evaporator were opened for inspection.

About fifty of the condenser tubes were found to have corroded; a myriad of small holes were present. Subsequent testing revealed that nodules of corrosion had formed on the inner tube surfaces (boiler makeup side). Dissimilar metals, principally iron and aluminum, had deposited on the surface forming differential aeration cells. This caused localized corrosion. It was determined that stainless steel tubes would correct the problem. At time of the report, new tubes had yet to be installed.

The evaporator was also examined; no corrosion was noted. However, the automatic cleaning system was nonfunctional, and about fifty brushes were lodged in their tubes, while the rest were either stuck in the brush cage or had entirely broken free. Because there was no sign of fouling, this cleaning method was not required.

ECONOMICS

The heat exchangers have operated more or less continuously since they were commissioned. The heat pump has operated intermittently during this period. The only prolonged performance was for a month long period before the condenser corrosion problem presented itself. The system has saved $1,100,000 in its first 20 months of operation, due almost entirely to savings generated by the heat exchangers, which were operational during the entire two-year study period. Based on an installed cost of $900,000, the heat pump system had a simple payback period of 1.4 years. The savings would have been increased (and the payback period shortened) had the heat pump been operating properly.

CONCLUSIONS AND RECOMMENDATIONS

A heat pump does not accommodate frequent stoppage or frequent variation in flows and temperatures. At time of installation, its flexibility was overestimated while variation of flow and temperature were underestimated.

For future installations, it is essential:

- to ensure a stable flow and temperature to the evaporator and condenser;
- to have good instrumentation for monitoring the operation by the plant personnel;
- to train personnel on the operation of new equipment; and
- to have an excellent line of communication with the heat pump manufacturer so that the machine specified is appropriate for the application.

LOBSTER MARKETING

This monitoring project was completed in February 1986 by Clearwater Lobsters Limited, in association with A. Orlic and Associates Limited, both of Halifax, Nova Scotia.

PROJECT RATIONALE

Traditionally a glut of lobsters occurs on the market in the spring and fall when the lobster fishing seasons are open, while a shortage of live quality lobsters exists during the balance of the year. Attempts to level out the supply cycle and fluctuation in market price in the past have failed because holding lobsters past the season has resulted in large waste through mortalities and quality reduction.

Experience within the industry has shown that during periods of cold ambient seawater, the mortality rate of the held lobsters was lower and reduction in quality was nominal.

In August, 1982, NRCC sponsored a feasibility study which identified the design and economics of a chilled saltwater lobster holding tank with condenser reject heat used for providing process and service water heating (see Lobster Marketing p.2-21).

In the fall of 1983, Clearwater Lobsters Limited commissioned a detailed design of the heat pump system. This led to the construction of the system and its functional operation in June 1984.

During the summer of 1984, the biological and energy performance results were adequately encouraging for Clearwater Lobsters Limited to initiate design and construction of a supplementary lobster holding facility. A warm seawater lobster holding laboratory was also added. All facilities were completed and commissioned by May, 1985.

The objective of this project was to determine the operating performance and economic benefit of the heat pump system.

SYSTEM DESIGN

The system consisted of a heat pump for cooling fresh seawater to 2 C. This seawater, when pumped into the lobster holding tanks, would maintain the tanks at 3 C. In order to minimize the capacity requirements of the heat pump, a heat exchanger was designed into the system to precool the incoming seawater with the 3 C water drained from the holding tanks (an equal amount of seawater must be drained from the holding tanks as is delivered by the heat pump). This configuration is shown in Figure 1.

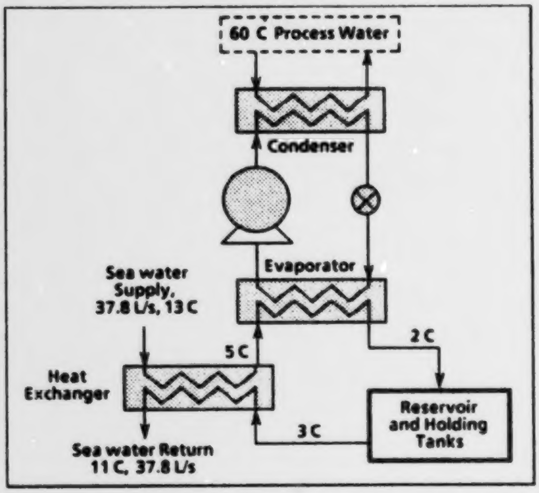

60 C Process Water
Condenser
Evaporator
Sea water Supply, 37.8 L/s, 13 C
5 C
2 C
Heat Exchanger
Reservoir and Holding Tanks
3 C
Sea water Return 11 C, 37.8 L/s

Fig. 1 Heat Pump Refrigeration Design for Clearwater Lobsters Ltd.

During parts of the spring and fall, when the ocean temperature is sufficiently cool, the heat exchanger would be bypassed and the heat pump would carry the full cooling load. During the winter both the heat exchanger and heat pump can be bypassed, with the cold ocean water supplying the entire cooling requirement of the holding tanks. However, since there are process water requirements, the heat pump is still operated, using the holding tank discharge water as the energy source.

Process water is supplied in excess of current requirements (cooling load is the determining factor in the heat pump utilization). Excess energy is dissipated through a heat exchanger on the process water loop using seawater .

SYSTEM MONITORING

The system was monitored for six months, from May to November 1985. During this entire period it operated with the heat exchanger in series with the heat pump. Monitoring included the continuous recording of temperature, flow rates and pressures at various locations in the system. Energy consumption of the heat pump and of the seawater pumps were also recorded.

PERFORMANCE

The system performed per design for only a very short period after commissioning. The fresh seawater side of the heat exchanger and evaporator steadily became fouled with marine growth, restricting flows to two-thirds to one-half of design values. Similar reductions in flow of the seawater discharged from the holding tanks were recorded, due to biological growth on the waste seawater side of the heat exchanger.

As a result, the system could not always meet the cooling requirements of the lobster holding tanks. The maximum design temperature in the holding tanks was 4.0 C, while the average temperature during the monitoring period was 4.8 C, with a maximum temperature of 6.9 C. One means of alleviating high temperatures for short periods was to shut off the fresh seawater intake and recirculate the holding tank seawater through the heat pump. During the summer one of the holding tanks was removed from the circuit, reducing the cooling requirement considerably.

There were some problems with the waste seawater discharge pumps and the seawater circulating pumps. These pumps exhibited very high rates of corrosion and required frequent maintenance and/or replacement of bearings and shafts. By contrast, the fresh seawater pumps were problem free.

ECONOMICS

The capital cost of the installation was $482,000.
The economic benefits from this installation are twofold. First, there are reduced costs for process water production using the heat pump. The second is improved lobster quality, including reduced shrinkage.

The process water produced by the heat pump was supplied at a cost of $36,000 (4.7¢/kWh) during the 6 month monitoring period. Doubling this gives an approximate annual cost of $72,000. Compared with generating this process water with an electric boiler, the total energy savings is $94,000. It should be noted, however, that not all of this process water is currently required and the above savings are not actually realized.

CONCLUSIONS AND RECOMMENDATIONS

The heat pump system could not maintain 4 C maximum design conditions, primarily due to fouling of the heat exchanger and evaporator. There were frequent utility power failures, which to a lesser extent were a contributing factor. Solutions to the heat exchanger fouling and pump corrosion require further investigation.

The benefits from the system have not yet been fully realized, either in terms of energy cost reduction or improved product quality. Product quality and shrinkage have improved significantly, but a number of biological factors and requirements remain to be investigated.

The heat pump system did however prove successful enough for Clearwater Lobsters to incorporate a heat pump system into a new lobster holding facility, which will soon be under construction.

DATA:

Process:	Supply process water
Waste heat source:	Sea water
Heat pump output:	417 kW
Working fluid:	R-12
COP:	2.36
Capital cost:	$482,000
Energy savings:	$94,000
Production savings:	Not determined

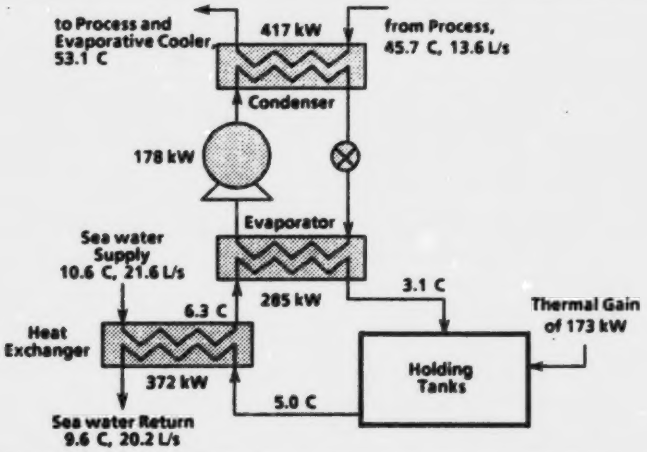

to Process and Evaporative Cooler, 53.1 C
417 kW
from Process, 45.7 C, 13.6 L/s
Condenser
178 kW
Evaporator
Sea water Supply 10.6 C, 21.6 L/s
285 kW
3.1 C
6.3 C
Heat Exchanger
Thermal Gain of 173 kW
372 kW
Holding Tanks
5.0 C
Sea water Return 9.6 C, 20.2 L/s

Fig. 2 Heat Pump Refrigeration Application at Clearwater Lobsters Limited.

Flow rates and temperatures, as well as heat transfer rates, are shown as average values during the monitoring period.

RESTAURANT - SERVICE WATER HEATING

E.J.Faraci and Associates, of Winnipeg, Manitoba, performed the following study in two parts; phases 'A' and 'B'. Phase 'A' was reported on p. 2-33. The following demonstration project (phase 'B') was carried out between September 1983 and November 1984.

PROJECT RATIONALE

The first phase of this heat pump application study had comprised the design and engineering evaluation of a heat pump wastewater heat recovery system at a Keg restaurant on Gary Street in Winnipeg. The design study had concluded that a heat pump system was feasible and would be cost effective.

This, the second phase of the study, demonstrates the operation of the system after installation in the restaurant.

The objectives of this project were:

- to install a heat pump system in a restaurant site;
- to monitor and evaluate heat pump system performance.

SYSTEM DESIGN

The heat pump system design was based on a water-to-water heat pump. This system extracted heat from from the dishwasher, refrigeration condenser and clothes washer wastewater streams and provided domestic hot water heating. A heat exchanger was placed in series with the heat pump (see Figure 1). During periods of heavy DHW demand, the water from the DHW preheat tank was heated by the heat exchanger and further heated by the heat pump. During periods of low demand, the heat exchanger was automatically bypassed, allowing the heat pump to operate alone.

OPERATING EXPERIENCE

The commissioning phase was burdened with equipment, control, and design related problems. A summary follows:

- an inadvertent connection of the prewash sinks to the sump, caused the sump level control to fail and twice the basement flooded;
- the first heat pump compressor would not operate due to a seized rotor. It was replaced;
- the input/output keyboard for the direct digital control system and the integrating energy meter failed;
- the second compressor operated at only 60 percent capacity. It was replaced;
- the third compressor performed well for 2 weeks until it abruptly failed. This was not replaced, and no further heat pump monitoring was performed;
- the wastewater pump became plugged and sludge built up inside the wastewater piping.

SYSTEM PERFORMANCE AND ECONOMICS

Prior to operation of the heat pump with the third compressor, the heat exchanger performance was monitored. Analysis of the data indicated the heat exchanger recovered only half as much energy as was originally predicted.

Once the third compressor was installed, the heat pump performance was monitored for 11 days with the heat exchanger bypassed. The heat pump output was 8 kW, 20 percent less than its rated capacity of 10 kW.

The system was then switched to operate with the heat exchanger and heat pump together. Data was recorded for 3 days before the third compressor failed. No further system monitoring occurred after this failure.

No explanation was noted in the study for the low capacity of the heat pump or the low recovery of the heat exchanger.

Based on the performance data obtained during the short monitoring period, annual cost savings of $1300 were estimated. This was significantly lower than the $3000 saving predicted in the design study, due primarily to the poor performance of the passive heat exchanger. The installed cost of the system was $35,000, substantially higher than the $15,000 predicted in the design study. Thus this research oriented heat pump system was not economically viable.

CONCLUSIONS

For a new restaurant, an installed cost for a heat pump system was predicted to be $8000, resulting in a simple payback period of 6 years, based on $1300 energy savings. Though technically feasible, the heat pump system will not be viable because of potential maintenance problems (fouling was detected in the wastewater circulation loop) and the relatively long 6 year payback period.

RECOMMENDATIONS

It was recommended that the Keg continue to operate their heat pump system, contingent on replacement of the failed compressor. Although the economic picture does not suggest that similar systems should be installed in other restaurants across the country, there was a positive cost benefit in maintaining the existing system. If evaporator fouling proves to be a maintenance burden, the heat pump should be used to only recover heat from the refrigeration equipment condenser cooling water.

The emphasis of future work should be directed towards the long term reliability and maintenance of the heat recovery system.

UPDATE

The failed compressor was replaced and the system continued to operate. No further monitoring was undertaken by NRCC.

DATA:

Process:	Service water heating
Waste heat source:	Restaurant wastewater
Refrigerant :	R-12
Heat pump output:	8.0 kW
COP:	4.1
Installed cost:	$ 8,000
Energy savings:	$ 1,300
Payback:	6 years

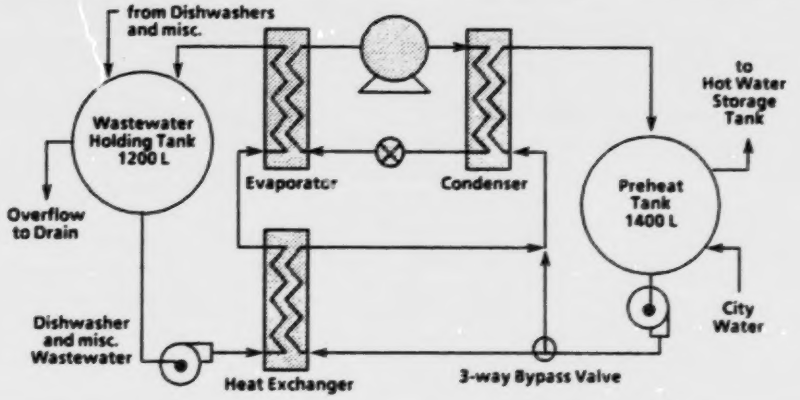

from Dishwashers and misc.
Wastewater Holding Tank 1200 L
Overflow to Drain
Evaporator
Condenser
to Hot Water Storage Tank
Preheat Tank 1400 L
City Water
Dishwasher and misc. Wastewater
Heat Exchanger
3-way Bypass Valve

Figure 1. Keg Restaurant Wastewater Heat Recovery System Utilizing a Heat Pump.

In the heat pump system illustrated above, a circulating pump on the wastewater side ensures a continuous flow of water first through the heat exchanger and then through the evaporator before it is returned to the wastewater holding tank.

Likewise a circulating pump on the DHW side ensures a continuous flow of water through the heat exchanger and the condenser. A three-way bypass valve is incorporated into the DHW circulation loop so that the heat exchanger can be bypassed, allowing the heat pump to operate alone.

During business hours, when hot water demand is high, city mains water is heated first at the heat exchanger. The heat exchanger ensures domestic water entering the condenser is always higher then the temperature of the wastewater entering the evaporator. If these conditions are not maintained, the compressor/motor may overload. The heat pump then upgrades this heat and provides preheated water to the storage tanks.

When little or no hot water is required, the water from the preheated storage tanks recirculates through the heat pump with the heat exchanger bypassed. Refrigeration equipment condenser cooling water provides the source of waste heat at this time.

A rotary compressor heat pump was chosen because it had a designed performance advantage over a reciprocating compressor when operating with high (eg. 30 C) evaporator temperatures. The commissioning period, however, had three compressors fail. This prevented extensive monitoring of the system and left concerns over the long term system reliability.

It was concluded that energy recovery from restaurant wastewater was technically feasible but that the economics do not support installations at this time.

COMMERCIAL LAUNDRY - SERVICE WATER HEATING

Hickling Partners Inc., of Ottawa, Ontario carried out this demonstration project between October 1982 and March 1985.

PROJECT RATIONALE

Commercial dry cleaning and laundering operations number approximately 4,000 across Canada and offer opportunities for novel applications of heat pumps. Heat pumps can recover heat from clothes washer wastewater and in turn provide domestic hot water for the washers.

The objectives of this study were to:

- select an appropriate commercial dry cleaning and laundering operation;
- identify the optimum size and performance of a heat pump waste recovery system for such an operation;
- select equipment, prepare an engineering design, and provide a statement of economic viability, benefits, and payback;
- install and operate the designed heat pump waste recovery system; and
- report on project findings including market penetration forecasts based on COP of this recovery system.

The work was completed in two phases. Planning and Design - phase one - dealt with the first three objectives. Objectives four and five were met in phase two - Commissioning and Operations.

PLANNING AND DESIGN

Betty Brite Cleaners at 218 Laurier Street East, in Ottawa, was selected because the firm provided maintenance and operating procedures affirming energy conservation, a staff supportive of the project and a continuous operation (6 hours per day, 6 days per week).

Current operating parameters were identified, factors such as laundry equipment operating cycles, flow rate and temperature of DHW delivered to the laundry equipment, flow rate and temperature of wastewater, and space limitations for the heat pump system.

The system configuration was developed and is shown in Figure 1.

A preliminary assessment of the system requirements led to the selection of a state-of-the-art Fedders packaged heat pump for the heat pump system. As a packaged system, it is equipped with a counterflow coaxial condenser and evaporator, a rotary compressor, factory installed thermal expansion valve and pre-wired controls. Subsequent computer analysis further indicated that only the largest Fedders model would be suitable (24 kW or 80,000 Btu/hr).

The system was modeled using a computer simulation program. This led to the selection of wastewater and DHW storage tanks each with capacities of 550 L. The only component which required special design was the wastewater storage tank; this was built on site.

Using Break Even Capital Cost calculations, an economic analysis was performed on the design. With oil costs of $0.323/L and electricity at $0.0426/kWh (1982), the payback period for the heat pump was calculated to be three years versus oil and five years compared to electrical heating. This assumed a predicted COP of 3.5 and installed cost of $7900.

COMMISSIONING AND OPERATIONS

The problems encountered during this period included:

- a three month delay in the delivery of the compressor;
- a condenser, attached backwards, had to be repiped. Plumbing specifications supplied by Fedders were incorrect;
- parallel piping of city and DHW water, which resulted in insufficient hot water for clothes washer. When connected in series, as intended, the problem was rectified;
- a failed compressor motor, which was replaced under warranty. The replacement was defective and was also replaced.

The maintenance factors identified were:

- strainers require cleaning every 2 weeks;

- wastewater storage tank requires drain-down every 4 weeks;
- the evaporator is predicted to need descaling once a year for optimum efficiency;
- heat pump pressures should be verified once a year by a qualified serviceman.

The results garnered during the operation of the system were:

- the heat pump system provided virtually 100% of the DHW requirements without the need for the heat exchanger;
- the average heat pump COP was 2.6 during the demonstration period;
- the installed cost of the system was $8000, and a payback period of 3.5 years was determined.

CONCLUSIONS AND RECOMMENDATIONS

The operation of a heat pump system at Betty Brite indicated a payback period of 3.5 years and a capital cost of $8000 (1982), based on three months operation. This compared closely to figures estimated prior to the demonstration (payback 3.0 years and capital cost of $7900).

The total Canada-wide energy reduction figure (0.1%) based on the assumed market penetration level of 50% by year 2000 is however, unsubstantiated and likely optimistic. First, dry cleaning and laundering operations with oil heating may find a less capital intensive conversion to gas heating more attractive. Second, the majority of establishments may be currently using cheaper sources, such as electricity or gas.

Though no recommendations were provided, on-going reviews of energy costs and maintenance requirements of the installed system would verify the 3.5 year predicted payback period. In addition, the regional economics of using the heat pump system in place of oil, gas and electric resistance heating should be explored, both for retrofit and replacement markets, in order to determine true market potential.

DATA:

Process:	Service water heating
Waste heat source:	Clothes washer wastewater
Operating hours:	36/week
Heat Pump output:	24kW
COP:	2.6
Installation cost:	$8000 (1982)
BECC payback:	3.5 years, compared to oil

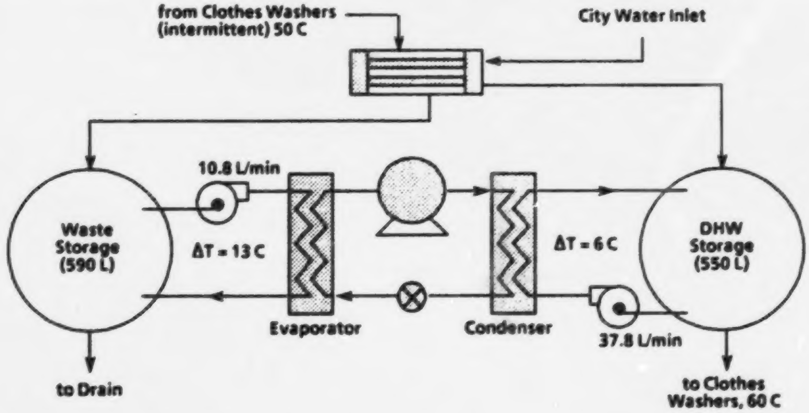


Figure 1. Service Water Heating at a Laundry and Dry Cleaning Establishment

In the above illustration, a heat exchanger is used to preheat city mains water, using clothes washer wastewater as the heat source. The wastewater then goes into a storage tank. The heat pump upgrades this heat, providing DHW.

All material selected are off-the-shelf items. No component required specific design for the installation except the piping. The heat pump was selected entirely on the basis of its heating capacity. The heat exchanger was selected for its ability to operate under laundry conditions, being readily serviceable and corrosion resistant.

It was concluded from the data analysis that the waste heat recovery system using only the heat pump without heat exchanger was capable of supplying all the necessary hot water requirements.

INDUSTRIAL REFRIGERANT COMPRESSION HEAT PUMP SYSTEM MODELLING

The following report concerns a heat pump computer simulation study carried out internally by the National Research Council of Canada, dated April 1985.

PROJECT RATIONALE

A simulation model for evaluating heat pump operation could prove beneficial in many instances:

- for site-specific engineering design of large industrial heat pump systems;
- to evaluate the performance of components and configurations when designing packaged units;
- to assist in the selection of packaged heat pump equipment and systems which would provide optimal performance for a specific commercial or industrial application.

The objective of this study was to simulate the operation of large closed cycle vapour compression heat pumps capable of process hot water or steam production.

SIMULATION PROCESS

The major components of a heat pump were identified in a "generic" model. This is shown in Figure 1. Equations were presented for each heat pump component. Together, they formed a system of equations requiring simultaneous solution, which when solved, simulated the operation of the various components of the heat pump system and the operation of the system as a whole.

The simulation required input to duplicate the conditions of the application - physical data, such as component sizes and configuration, and variable data, such as heat source and sink environments, Refrigerant thermodynamics and manufacturer's published compressor performance specifications.

CONCLUSIONS AND RECOMMENDATIONS

A simulation technique for comparing heat pumps on the basis of overall performance, such as COP, was developed. It was designed for eventual computer application.

The simulation should be validated with data and performance evaluations obtained from heat pumps already in industrial operation.

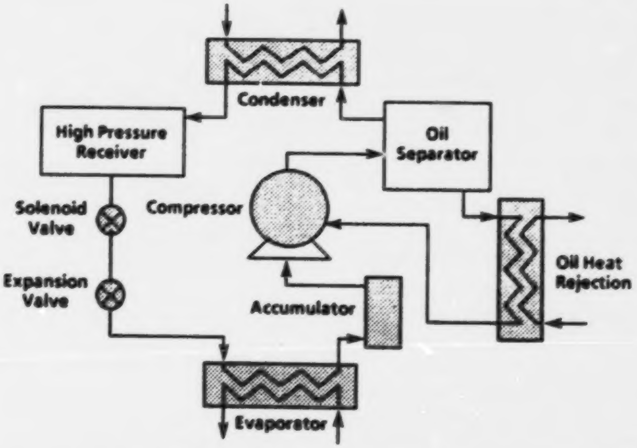


Figure 1. Schematic of a Simulated Closed Cycle Vapour Compression Heat Pump.

The generic heat pump illustrated above is used for hot water or steam generation. Major components include a compressor, shell-and-tube condenser, shell-and-tube evaporator and refrigerant expansion device. The refrigerant completes a unitary cycle through the shellside of each heat exchanger.

SPACE HEATING WITH THE HEAT PUMP

The National Research Council of Canada, Ottawa, published the following report in January 1978.

PROJECT RATIONALE

This publication was prepared to provide an overview of air-source heat pump performance, economics and possible design improvements for residential heat pumps. Heat pump systems utilizing alternative heat sources were also explored and reported on.

HEAT PUMP PERFORMANCE

The term 'balance point' is explained as being the outdoor temperature at which the heat pump can just satisfy the heating requirement of a conditioned space. At outdoor temperatures below the balance point the difference between the heat pump's capacity and the building heat loss has to be made up by auxiliary heaters, resulting in a further reduction in the unit's seasonal 'Coefficient of Performance' (COP). At outdoor temperatures above the balance point the heat pump has excess capacity, and so must be cycled on and off in order to avoid overheating the conditioned space.

The 'Seasonal Performance Factor' (SPF), or average heat pump system efficiency over an entire heating season, was said to be dependent upon: 1) the unit's operating characteristics such as capacity, power requirement and COP vs. outdoor temperature; 2) the building's heat loss characteristics; and 3) the frequency and severity of outdoor temperatures at which space heating is required.

Calculations for an Ottawa location indicate that an all-electric heat pump (with auxiliary electrical resistance heaters) would reduce heating energy use by 37% over straight electrical resistance heating.

POTENTIAL DESIGN IMPROVEMENTS

Three design areas identified to improve heat pump season performance were: 1) capacity modulation by varying the compressor speed during operation at outdoor temperatures above the balance point; 2) reduction of outdoor coil icing during cool, humid weather by special coil design; 3) relocation of the components, i.e. compressor and reversing valve, from an outdoor to an indoor location in order to make use of their heat losses during the heating season.

ALTERNATIVES FOR AUXILIARY HEATING

Residential heat pumps were generally used in combination with another (auxiliary) heating system, such as an electric furnace. A suggested alternative to this all-electric configuration was the use of fossil fuelled auxiliary heaters. The significant disadvantage of this system was that the furnace and heat pump could not be operated simultaneously below the balance point. This is because the heat pump indoor coil must be located downstream from the furnace heat exchanger to avoid heat exchanger corrosion due to condensation forming on the inside of the furnace heat exchanger during the cooling season. With the indoor coil of the heat pump downstream from the furnace, simultaneous operation of the heat pump and furnace would result in high compressor power draw and refrigerant discharge temperatures in the heat pump, adversely affecting heat pump performance.

ECONOMICS

The economics of air source heat pump installations were discussed. Several different combinations of circumstances were examined, and maximum justifiable investments were calculated for each case. The report pointed out the importance of future trends in electricity cost (and interest rates) to the economic viability of a heat pump installation.

ALTERNATIVE HEAT SOURCES

The report describes several alternatives to using air as the heat source, including subsurface ground coil systems, solar-assisted heat pumps and ice-maker heat pump systems (see Figure 1).

A ground coil system installed in a Toronto home and monitored over three heating seasons (1949-1952) provided 43 percent of the residence's heating requirement. The size and configuration of heat exchanger used in a ground coil system depends largely upon the depth of the water table. Where summer cooling is necessary, the ground is a poor heat sink. The heat dissipated from the conditioned space through the coil causes moisture migration in the soil, and voids and cavities around the coil result which inhibit heat transfer. Although installation cost was cited as another limiting factor, ground source heat pumps would possibly offer performance advantages over air source heat pumps which might offset the greater capital investment required for the ground coil.

Solar assisted heat pump systems were cited as another alternative. If some form of thermal energy storage is employed, the collection problems due to the intermittent nature of solar radiation can be largely overcome. The effectiveness of the solar assisted heat pump is governed by the collector area and thermal storage capacity. Increasing the size of the collector area (for a fixed storage capacity) seems to have more effect on system performance than increasing the storage (with the collector area fixed). Collector and storage area are subject to cost and space limitations, however, and for this reason other options deserve consideration. For example, the provision of an air-to-air heat pump mode for mild, sunny days (when solar assist is not needed) would free the collector loop to recharge the storage tank(s). Under mild, overcast conditions, this mode of operation would still satisfy the heating requirements without depleting the thermal storage. A system with storage capacity on both the condenser and evaporator sides of the heat pump would offer performance advantages over a single storage system, as it would be possible to use a smaller capacity heat pump operating almost continuously, although not necessarily meeting the heating requirements at any given time.

Oak Ridge National Laboratory had been experimenting with the development of 'Ice-maker' heat pumps. An aluminum plate evaporator replaces the conventional cross flow air-to-refrigerant heat exchanger of a typical air-to-air heat pump. A circulator delivers chilled water to a distributor located above the evaporator plate. The water freezes on the surface of the evaporator which is maintained at approximately -7 C (20 F), liberating the latent heat of fusion to the evaporating refrigerant.

The report indicates that further investigation into these three alternatives is required in order to assess their potential for residential application.

CONCLUSIONS

It was concluded that air-to-air heat pumps are able to compete economically with electrical resistance heating systems.

Three important factors which must be considered when discussing alternative heat sources for residential heat pump systems were pointed out. These were the initial cost to the consumer, the reliability of the system, and the load factor and effects on the utility. (One alternative system not discussed in this report is the well-water heat pump system. These heat pumps were investigated in subsequent NRCC reports.)

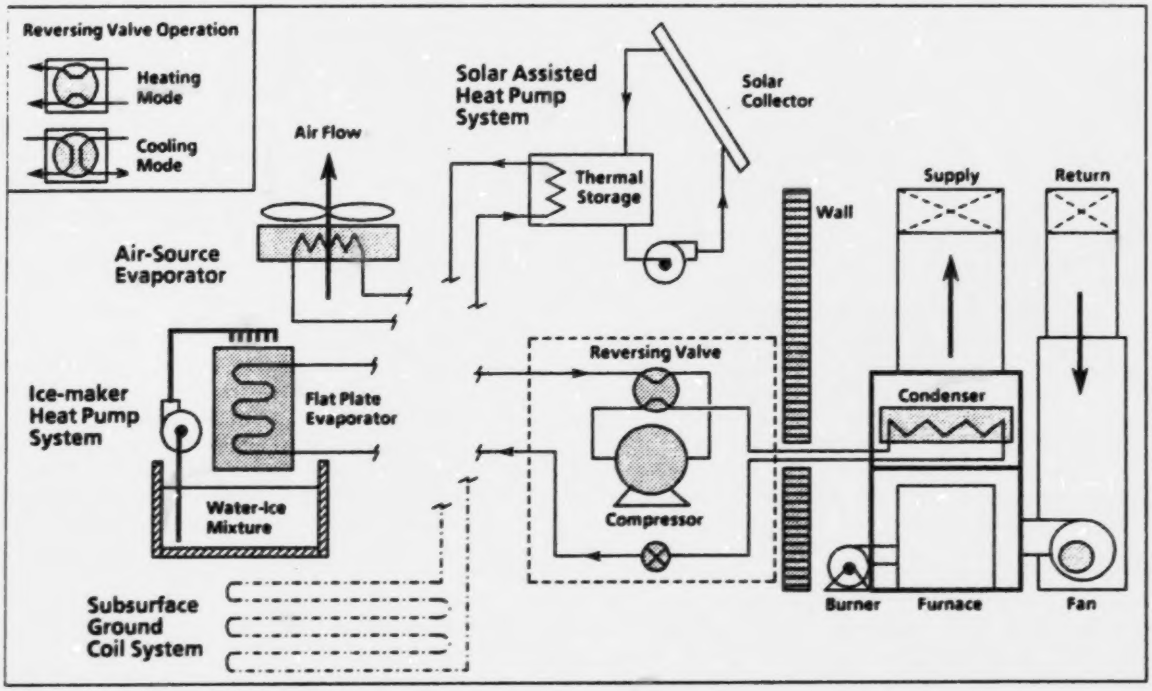

Reversing Valve Operation
Heating Mode
Cooling Mode
Solar Assisted Heat Pump System
Solar Collector
Air Flow
Thermal Storage
Supply
Return
Wall
Air-Source Evaporator
Reversing Valve
Ice-maker Heat Pump System
Flat Plate Evaporator
Condenser
Water-Ice Mixture
Compressor
Subsurface Ground Coil System
Burner
Furnace
Fan

Figure 1. Several Configurations of a Residential Heat Pump Heating System

THE COST OF OWNING AND OPERATING A HEAT PUMP

This report was prepared by the National Research Council of Canada, dated April 1980.

PROJECT RATIONALE

The objective of this study was to outline the operational characteristics of various residential heat pumps and compare the (1979) costs of owning and operating such equipment with conventional furnaces. The comparison was intended to allow the consumer to select an appropriate heating system, either for retrofit, replacement, or for new construction, on the basis of life cycle cost.

HEATING SYSTEM PERFORMANCE

The heating energy requirements of a 'typical' house located in various cities across Canada were calculated, using a method developed for the study. The performance of the various heating systems under consideration were reviewed, and Seasonal Utilization Efficiency (SUE) was introduced; the SUE for electric furnaces being 100% and for heat pumps usually 100-200%. Fossil fuelled systems, which are dependent on load, are considerably less than 100%.

The different types of heat pump systems were described briefly (see Figure 1), along with the associated advantages and disadvantages of each. These systems were grouped into the following categories:

- All-electric heat pump: air-to-air heat pump equipped with supplementary electric heating elements;
- Add-on heat pump: air-to-air heat pump that is retrofitted to a fossil fuelled furnace, with the indoor coil placed downstream of the furnace;
- Hybrid heat pump: air-to-air heat pump added to a fossil fuelled furnace, with the indoor coil located upstream from the furnace. This system is capable of continuous operation below the balance point.

ECONOMICS

Energy costs and fuels availability for the various regions were tabulated, and annual heating energy cost graphs for the various systems were prepared for each city considered in the comparison. Capital and annual maintenance costs were presented for each system referenced on figures for Ottawa. Special costs such as upgraded electrical service or chimney improvements were included, and total life cycle costs were generated and presented graphically. The life cycle costs were determined by calculating the present worth of energy and maintenance costs for the life of the system and adding them to the system's capital costs.

CONCLUSIONS

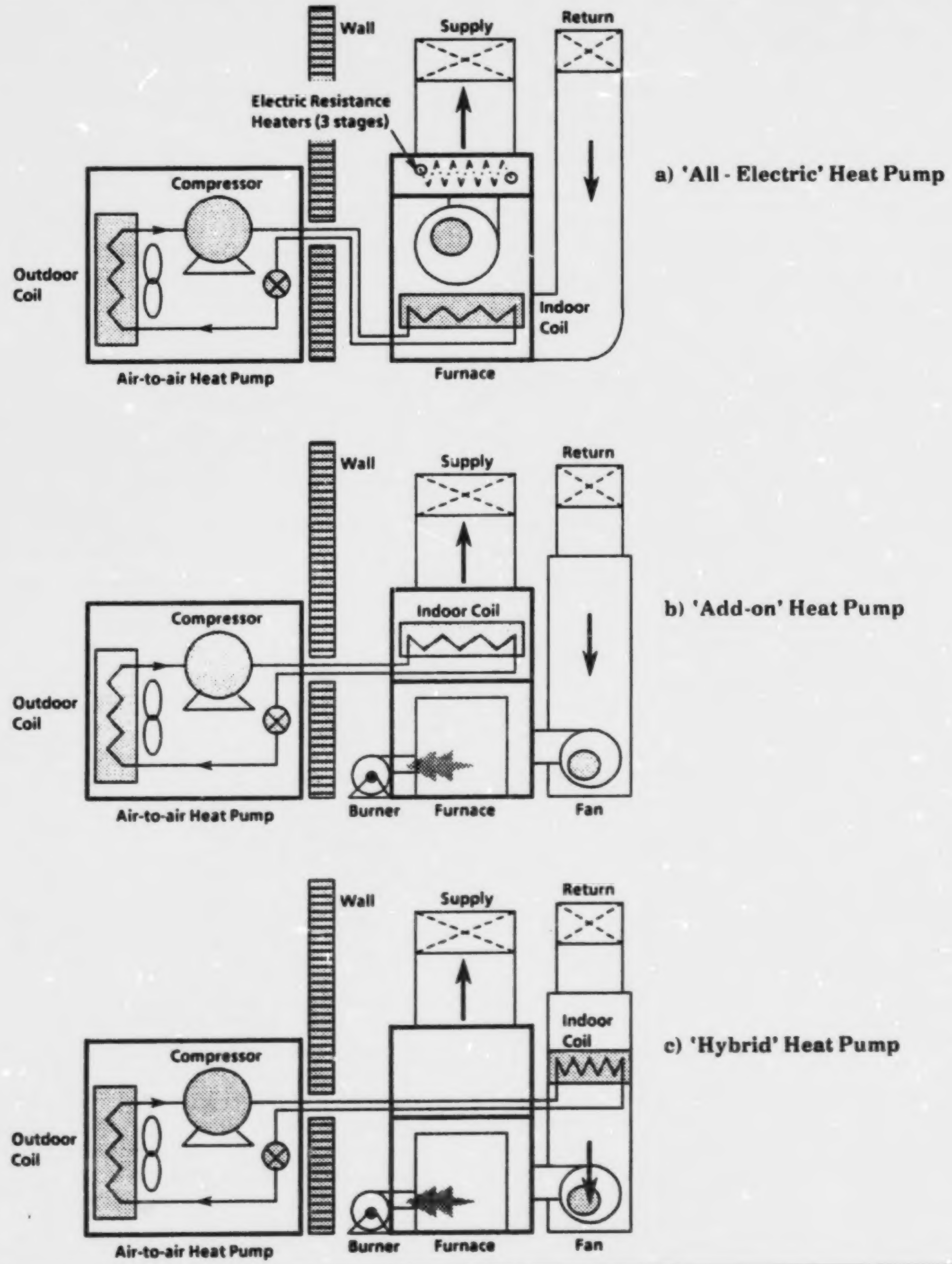
Based on certain assumptions made in the report, it was determined at that time that a natural gas furnace was the most economical choice for replacement of oil furnace heating systems, new construction, or retrofit applications. The assumptions on heat pump service life (12 years) and future trends in energy prices and interest rates were in retrospect conservative.

In areas where natural gas was not available, and depending on the annual fuel consumption, an electric furnace or a heat pump system would be the preferred choice for new construction in all regions except Halifax, N.S.

In instances where natural gas was not available and an existing oil furnace needed replacement, an oil-hybrid heat pump would often be justified, and without a capital cost allowance for air conditioning. However, where an existing oil furnace had a useful service life remaining, an air conditioning capital cost allowance must be made to justify an add-on heat pump. Although add-on heat pumps which allow operation below the balance point were not included in this analysis, they would in most cases be an economical retrofit without an air conditioning capital allowance.

RECOMMENDATIONS

It was recommended that the homeowner be aware of local capital and maintenance costs. It was stressed that these costs vary considerably from one region to another, and that the figures used to generate the 'Total Life Cycle' costs were based on prices quoted in the Ottawa area.


Wall
Supply
Return
Electric Resistance Heaters (3 stages)
Compressor
Outdoor Coil
Indoor Coil
Air-to-air Heat Pump
Furnace
Indoor Coil
Burner
Fan

a) 'All - Electric' Heat Pump

b) 'Add-on' Heat Pump

c) 'Hybrid' Heat Pump

Figure 1. Three Residential Heat Pump Systems - Heating Mode shown

THERMAL AND ECONOMIC PERFORMANCE OF TEN SPACE HEATING SYSTEMS

Spider Engineering Associates of Waterloo, Ontario, undertook this study under contract to the National Research Council of Canada between 1982 and 1984.

PROJECT RATIONALE

Recent interest in efficient energy utilization had led to the development of a number of new residential space heating systems. As a result consumers were faced with an unprecedented number of options when selecting a heating system.

The objective of this work was to compare the thermal and economic performance of the various heating systems in different regions of Canada, for both new and existing homes, and thus provide the consumer with a basis for making a selection.

APPROACH

A total of ten heating systems were evaluated for eight Canadian cities for both new and existing housing applications. Two computer simulation programs were employed: HEAT to model the performance of the different heating systems and WATCORE to estimate the hourly heating demand for two identical houses with different levels of insulation and air tightness. Hourly weather data, for each city, was obtained from weather tapes developed by NRCC.

Economic performance was calculated for different economic indices: life cycle costs, discounted payback period, internal rate of return and first year energy costs.

Four of the ten systems evaluated were heat pump systems; a well-water heat pump; an all electric air-source heat pump; an all electric 2-compressor air-source heat pump and an add-on heat pump. There was no capital cost allowance for the air conditioning capability of the heat pumps nor was the cost of central air conditioning added to the cost of the alternative heating systems.

The other heating systems evaluated were conventional oil and gas furnaces; medium efficiency and condensing gas furnaces; electric resistance furnace and the electric plenum heater.

RESULTS

Heating system performance and economics are presented in tables similar to Table 1 and 2 respectively for each of the eight Canadian cities.

In addition, the resulting impact of the following factors on heating system performance are presented and discussed:

Table 1. Annual Heating System Performance for Toronto

House Model	Annual Heating Demand (MJ)	Heating System	Annual Energy Consumption		
			GAS (m^3)	OIL (L)	ELECTRICITY (kWh)*
House 1	40,132	Conventional Gas	2,067	0	465
		Conventional Oil	0	1,839	1,608
		Medium Efficiency Gas	1,643	0	672
		Condensing Gas	1,167	0	709
		Electric Forced Air	0	0	11,309
		Water-Source Heat Pump	0	0	4,201
		Air-Source Heat Pump	0	0	5,011 / 1,439
		2 - Compressor Heat Pump	0	0	5,954 / 263
House 2	66,087	Conventional Gas	3,033	0	683
		Conventional Oil	0	2,765	2,084
		Medium Efficiency Gas	2,604	0	1,033
		Condensing Gas	1,921	0	1,126
		Electric Forced Air	0	0	18,623
		Water-Source Heat Pump	0	0	6,723
		Air-Source Heat Pump	0	0	6,946 / 4,012
		2 - Compressor Heat Pump	0	0	9,077 / 1,133
		Add - on Heat Pump	0	899	7,149
		Electric Plenum Heater	0	1,819	9,261

*For double entry x/y, x represents the heat pump consumption, and y represents the auxiliary heater consumption.

- furnace oversizing
- heat pump capacity
- heat pump control method
- plenum heater control method

The assumptions or basis for energy prices, service life, discount and inflation rates, equipment costs, maintenance and repair costs are also presented.

CONCLUSIONS

The following conclusions were drawn regarding the performance of the various heating systems:

- the addition of a stack damper and electronic ignition to a conventional natural draft gas furnace can improve seasonal efficiency by approximately 15 to 25 percent;
- a condensing gas furnace can achieve a 60 percent increase in seasonal efficiency compared to a conventional natural draft gas furnace;
- proper sizing of a conventional gas furnace resulted in a seasonal efficiency improvement of six percentage points;
- all-electric air-source heat pump predicted energy savings ranged from a low of 25 percent in Winnipeg to a high of 55 percent in Vancouver, in comparison to electric resistance heating;
- a two-compressor heat pump increased the energy savings achieved by a single compressor heat pump by anywhere from 8 to 12 percent depending on location.

The following conclusions were drawn regarding the economics of the various heating systems:

- For *heating only* applications, in new construction, the lowest life cycle costs and payback periods were associated with condensing and medium efficiency gas furnaces;
- For *heating only* applications, in existing housing, the condensing gas furnace and well-water heat pump exhibited the lowest life cycle costs, depending on location.

RECOMMENDATIONS

It was recommended that the models of the add-on heat pump and electric plenum heater required further work as agreement between predicted and actual field observations of performance were significantly different (i.e. field performance better than predicted by model). This was attributed to the complexity of the furnace chimney flow and house heating load interaction.

Though not recommended in the report, the economic analysis should be repeated to provide the most cost-effective combined heating/cooling system, as central air conditioning is increasingly popular in certain regions of Canada.

Table 2. Economic Results for Toronto

House Model	Heating System	Life Cycle Costs		Discounted Payback Period		Internal Rate of Return		First Year Energy Costs	
		$	Rank	Years	Rank	%	Rank	$	Rank
House 1	Conventional Gas	7,325	3	-	-	-	-	452	6
	Conventional Oil	10,391	8	>15		<1		687	8
	Medium Efficiency Gas	6,639	2	3	1	56	1	372	5
	Condensing Gas	6,167	1	6	2	31	2	275	3
	Electric Forced Air	7,752	4	>15		<1		498	7
	Water-Source Heat Pump	9,524	6	>15		3	3	185	1
	Air-Source Heat Pump	8,889	5	>15		<1		284	4
	2 - Compressor Heat Pump	9,924	7	>15		<1		274	2
House 2	Conventional Gas	8,419	3	4	1	38	2	663	7
	Conventional Oil	10,977	8	-	-	-	-	1,018	10
	Medium Efficiency Gas	7,670	2	5	3	36	3	589	5
	Condensing Gas	6,637	1	4	1	42	1	450	3
	Electric Forced Air	10,362	6	12	6	17	5	819	8
	Water-Source Heat Pump	8,601	4	9	4	20	4	296	1
	Air-Source Heat Pump	10,268	5	14	7	15	6	482	4
	2 - Compressor Heat Pump	11,091	9	>15		12	8	449	2
	Add - on Heat Pump	10,542	7	11	5	15	6	616	6
	Electric Plenum Heater	11,590	10	>15		<1		1,017	9

ADD - ON HEAT PUMP PERFORMANCE AND SELECTION

This paper was published by the National Research Council of Canada in October 1981.

PROJECT RATIONALE

As a means of reducing oil consumption, the Government of Canada had initiated a program that offered a grant to homeowners who convert their existing oil heating system to other forms of energy such as natural gas or electricity. This grant also applied to add-on heat pump systems, if this addition reduced oil consumption to 50 percent of previous annual levels.

The addition of an electric heat pump to an existing oil furnace will, in most cases, reduce annual oil consumption by more than 50 percent. The actual saving in energy costs will depend on the amount of electricity used by the heat pump, and oil and electricity costs.

The objective of this report was to supply homeowners with a simple calculation method to determine the electricity and oil consumption that would result from adding a heat pump to their present furnace.

DESIGN

The heat pump provides all the heating needs of the home until the heat loss from the house exceeds the capacity of the heat pump. This typically occurs in the vicinity of -5 C to 0 C and is called the balance point. When temperatures drop below the balance point, the system operates in one of two ways:

(a) the heat pump shuts off and the furnace provides all the heating; or

(b) the furnace and the heat pump share the heating duty, but never operate simultaneously.

With either method (a) or (b), primary control is by a central indoor thermostat. Thermostat controls located in the furnace, in case (b) and an outdoor thermostat, in both (a) and (b), control the operation of the heat pump. An add-on heat pump system is shown in Figure 1.

CALCULATION METHOD

Charts are presented in the report for eight Canadian cities, permitting the homeowner to determine the expected oil and electricity consumption after an add-on heat pump is fitted to the furnace. The only information required to use the charts is the home's historic heating oil consumption, the heat pump size, and the control method. Manufacturers will provide an estimate of the size required (based on the cooling capacity) and the method of control, (a) or (b). A sample chart and an example of how to use it are presented in Figure 2. In that example an Ottawa homeowner can

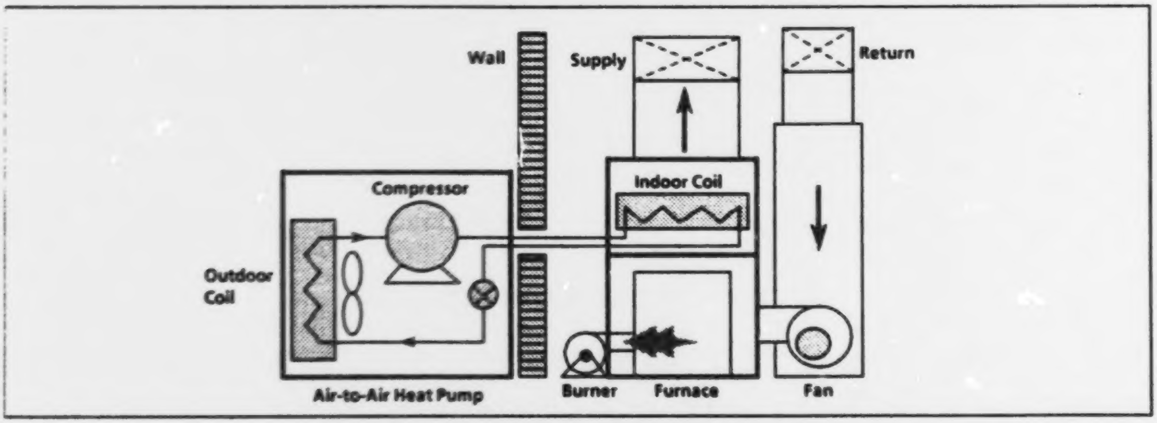


Figure 1. Add-On Heat Pump System - Heating Mode Shown

Figure 1 (above) illustrates an add-on heat pump used to supplement oil furnace heating. The heat pump supplies all heating requirements to the balance point, which normally occurs at -5 C to 0 C. At ambient temperatures below the balance point, the heat pump either shuts off or shares the heating duty with the furnace, depending on the control method employed. At no time do the furnace and the heat pump operate simultaneously.

realize a net annual savings of $345 by converting to an add-on heat pump.

Other factors affect the cost of owning and operating the add-on heat pump system, including the cost of service or preventative maintenance.

CONCLUSION

In most cases, an add-on heat pump provides at least 50 percent reduction in oil consumption, an eligibility requirement for government offered off-oil conversion grants.

UPDATE

The government grant was discontinued in 1985.

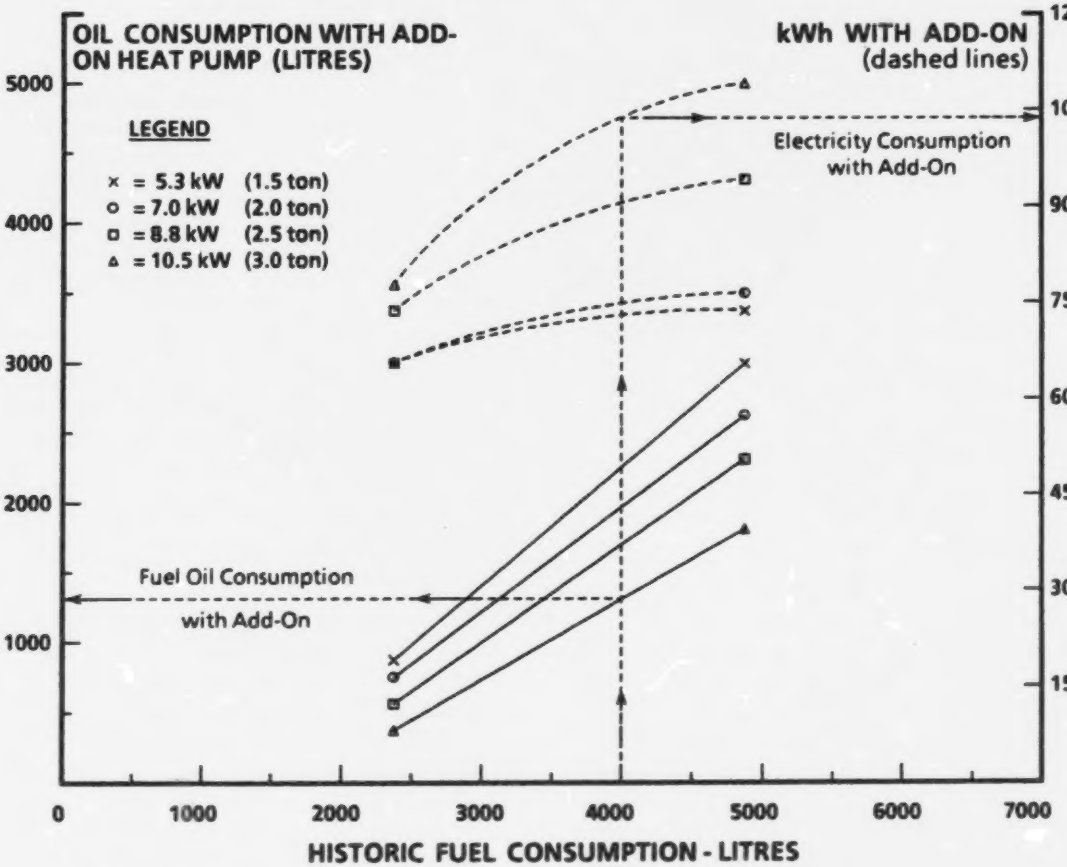


Figure 2. Ottawa: Add-on Heat Pump with Operation Below the Balance Point

The above graph allows a homeowner to predict oil displacement resulting from the installation of an add-on heat pump. For example:

Annual Average Oil Consumption	**- 4000 litres**
Annual Average Oil Consumption with Add-On Heat Pump (10.5 kW / 3 ton)	**- 1330 litres**
Reduction in oil consumption	**- 67%**
Electrical consumption	**- 10,350 kWh**

With a heating oil cost of 25¢ / litre and electricity cost of 3.12¢ / kWh, savings can be calculated as follows: (4000 - 1330) x 0.25 - (10,350 x 0.0312) = $345

CALCULATING HEAT PUMP ENERGY REQUIREMENTS

National Research Council of Canada published the following paper on heat pump energy use calculation September, 1979.

PROJECT RATIONALE

Air-source heat pump purchased energy requirements could be calculated using the ASHRAE bin method presented in the ASHRAE Systems Handbook (1976). This method was thought to over-estimate the heating energy requirements and thus exaggerate the heating energy savings of the heat pump.

This report presents an alternative to the ASHRAE bin method that would more accurately predict the space heating load and heating energy savings.

APPROACH

Three refinements were introduced to modify the ASHRAE bin method. First, the effects of internal and solar heat gains were included. Second, a Seasonal Heat Loss Reduction Factor (SHLRF) for each building component was introduced. This makes heat loss calculation for the structure more accurate than the single overall degree-day correction factor used in the ASHRAE bin method. Finally, a "break even" outside air temperature was established above which internal and solar gains offset building heat loss.

Infiltration rates were assumed to range between 0.2 and 0.5 air changes per hour. Heat loss from below grade surfaces were assumed to be constant and independent of indoor-outdoor temperature differences.

Solar heat gains through windows were estimated using factors derived from NRCC research. Internal heat gains included all electrical energy consumed by appliances and lights, plus 60 percent of the energy used for domestic hot water, plus heat generated by the house's occupants (estimated on an average occupancy of 2, for 12 hours per day at 65 watts per occupant). All figures were applied to a heating season defined as all months with a mean temperature less than the "break even" temperature.

RESULTS

An existing two-story single detached house (heated floor area 164 m^2) in Ottawa (4673 Celsius degree-days) was used to illustrate the application of the "modified" bin method. First calculations were made using the "modified" bin method; then the ASHRAE bin method; and finally both results were compared to field monitored data from the same house. The results are summarized below and illustrated in Figure 1.

"Modified" Bin Method

Total seasonal heating energy	11,520 kWh
Seasonal Performance Factor	1.9
Energy saving over resistance heating	5,340 kWh

"ASHRAE" Bin Method

Total seasonal heating energy	20,070 kWh
Seasonal Performance Factor	2.2
Energy saving over resistance heating	11,120 kWh

Monitored Results

Total seasonal heating energy	10,500 kWh

Note that the Seasonal Performance Factor, which was calculated separately using the "Modified" Bin Method and the ASHRAE Bin Method, is higher when using the latter method. This is because the ASHRAE Bin Method overestimates heat pump use during mild weather when the heat pump operates at high COP's.

CONCLUSIONS

The "modified" bin method prediction of total heating energy required proved to be much closer to actual energy use than the ASHRAE method; 11,520 kWh "modified" estimate versus 20,070 kWh ASHRAE estimate on a house that was subsequently observed to need 10,500 kWh.

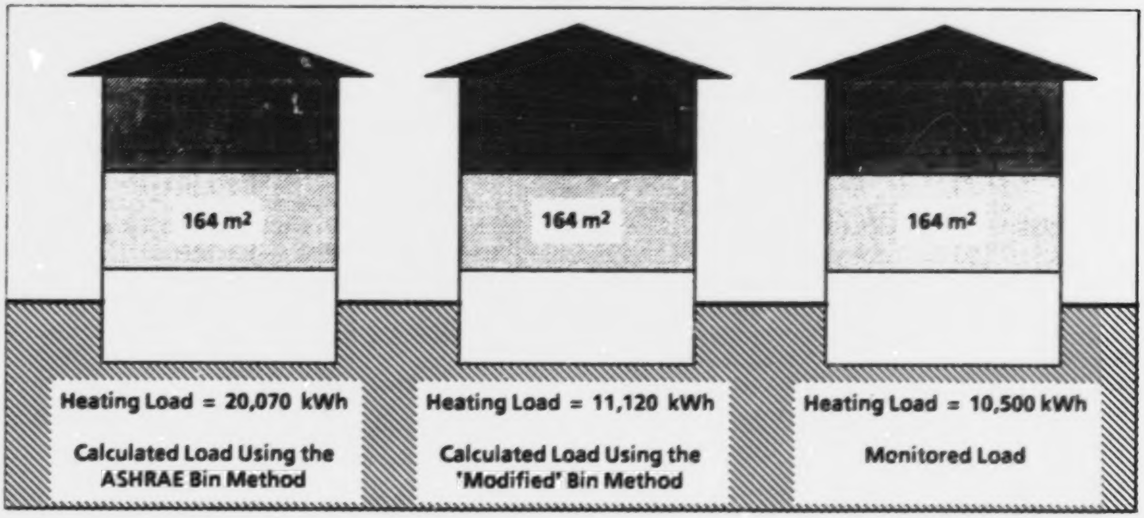

164 m²
164 m²
164 m²
Heating Load = 20,070 kWh
Calculated Load Using the ASHRAE Bin Method
Heating Load = 11,120 kWh
Calculated Load Using the 'Modified' Bin Method
Heating Load = 10,500 kWh
Monitored Load

Figure 1. Variations Between Calculated Heating Loads and Comparison with Monitored Load

HEAT PUMPS FOR HYDRONICALLY HEATED HOMES

The Ontario Research Foundation performed this feasibility study, dated May 1984.

PROJECT RATIONALE

Add-on heat pumps are normally installed in homes which already have forced-air distribution systems. It is possible, however, to apply add-on heat pumps to homes with hydronic (hot water) distribution systems, if air-to-water heat pumps are used.

The objective of this study is to evaluate the technical and economic feasibility of retrofitting heat pumps to homes with hydronic distribution systems.

DESIGN AND METHOD OF EVALUATION

An air-to-water heat pump would be coupled to an existing hydronic heating system. Return water from the hydronic distribution system would first pass through the heat pump and then the boiler. During periods of mild weather, the heat pump would supply the entire heating requirement, while in periods of colder weather the boiler would supplement the heat supplied by the heat pump. A schematic of the system is shown in Figure 1.

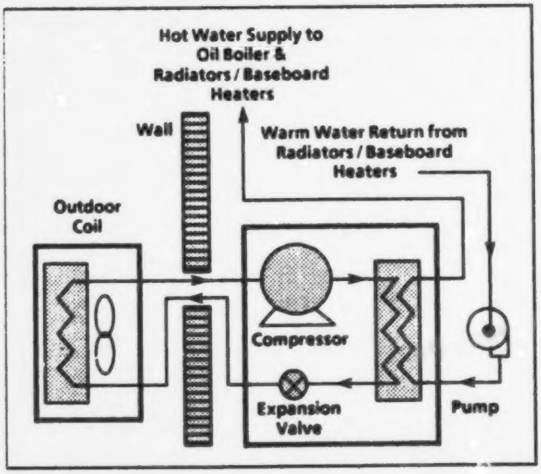

Hot Water Supply to Oil Boiler & Radiators / Baseboard Heaters
Wall
Warm Water Return from Radiators / Baseboard Heaters
Outdoor Coil
Compressor
Expansion Valve
Pump

Figure 1. Air - to - Water Heat Pump for Hydronic Heating Application. Summer Cooling is not Provided.

The approach taken was to prepare a computer model of the hydronic heating system with the add-on heat pump and simulate its performance.

Two Carrier air-to-water heat pumps, of 11 kW and 16 kW nominal capacities, were selected for the analysis. Manufacturers data was used to model their performance.

Baseboard heating units were tested in the laboratory using Ontario Research Foundation's warm-wall test booth. The tests determined the heat transfer characteristics of the baseboard heating units over a wide range of supply water temperatures. Three commonly used units were tested: a cast iron baseboard, an aluminum fin baseboard; and a cast iron radiator. The performance data was incorporated into the computer simulation of the hydronic heat pump systems.

ECONOMICS

The computer simulation predicted the purchased energy used by the hydronic heat pump systems. Life cycle costs and discounted payback periods were determined, based on conversion costs and reduced operating costs. The economic analysis was performed for houses with five different heating load characteristics in three different locations (Halifax, Montreal and Toronto) for different heat pump sizes and hydronic distribution systems. Life cycle costs and discounted payback periods were also determined for three other conversion options. These were for conversion to electric and natural gas boilers, and for conversion of an oil burner to a natural gas burner in existing boilers. A sample of the results of this analysis is presented in Table 1, for a home with an historic oil consumption of 3900 L/yr and a heat pump with a capacity of 16 kW (8.3 C outdoor air, 38 C return water temperature).

Oil displacement generally ranged from 70 to 80 percent, and life cycle costing showed that there was a net saving for those homes which consumed greater than 3500 L of oil per year. For the homes presented in Table 1, discounted payback periods were in the range of 9 to 13 years. However, for Montreal and Toronto where natural gas is available, replacement of an oil burner with a natural gas burner yielded significantly reduced life cycle costs and payback periods.

OTHER DESIGN OPTIONS

Laboratory tests were conducted using tangential blower assist on the baseboards and radiator.

Table 1. Economic Analysis of 16 kW Hydronic Heat Pump Conversion in a Home with an Historic Oil Consumption of 3900 L / year (All dollars are in 1984 dollars)

Conversion Type	Installed Cost ($)	Operating Cost[1] ($)			Percent Oil Displacement (%)			Life Cycle Cost ($)			Life Cycle Difference in Cost ($)			Discounted Payback Period (yrs)		
		Hfx	Mtl	Tor	Hfx	Mtl	Tor	Hfx	Mtl	Tor	Hfx	Mtl	Tor	Hfx	Mtl	Tor
Oil Boiler (Base Case)	-	1420	1460	1460	-	-	-	16200	16500	16600	-	-	-	-	-	-
Heat Pump & Aluminum Fin Baseboard	4600	1070	940	910	79	68	73	15500	14700	14600	700	1800	2000	12.9	9.8	9.7
Heat Pump & Cast Iron Radiator	4600	1060	910	870	85	71	79	15300	14400	14100	900	2100	2500	12.3	9.2	9.0
Heat Pump & Cast Iron Baseboard	4600	1050	890	860	87	73	80	15200	14100	13900	1000	2400	2600	11.9	8.8	8.7
Electric Boiler	2700	1690	1060	1130	100	100	100	19500	13900	15200	none	2600	1400	N/A	6.7	9.9
Gas Boiler	2520	N/A	1030	880	N/A	100	100	N/A	14500	12800	N/A	2000	3800	N/A	6.8	4.8
Conversion Gas Burner	1170	N/A	1020	870	N/A	100	100	N/A	13000	11300	N/A	3500	5300	N/A	2.8	2.0

[1] Includes energy costs and maintenance.
Halifax: Oil, 34.8¢/L; Electricity, 6.80¢/kWh; Natural Gas, not available
Montreal: Oil, 35.2¢/L; Electricity, 3.95¢/kWh; Natural Gas, 24.2¢/m^3
Toronto: Oil, 36.3¢/L; Electricity, 4.52¢/kWh; Natural Gas, 21.2¢/m^3

effectively reducing the return water temperature and increasing heat pump utilization. The computer simulation results for this option showed that oil displacement was increased by an additional 10 percent.

Through-the-wall 'packaged terminal' heat pumps were evaluated as an alternative retrofit to hydronically heated homes. These heat pumps carried all of the heating load down to 2 C, at which time the existing oil boiler assumes the entire heating load. Conversion costs were comparable to those for hydronic heat pump conversions. Using computer simulations similar to those used for the hydronic heat pump system previously described, oil displacement was found to range from 34 percent in Montreal to 42 percent in Halifax.

CONCLUSIONS AND RECOMMENDATIONS

The hydronic industry was found to have oversized baseboard capacity by a factor of approximately two. The significance of this oversizing is that a large portion of the total heating load can be carried by the baseboard units at the lower temperatures characteristic of heat pump hydronic systems, i.e. maximum of about 50 C.

When the economic performance of the various conversion options was examined, it was found that the gas conversion burner was generally the best option. However, this option is limited by the availability of natural gas and the original boiler's suitability for the conversion. If the gas conversion burner is not an available option the heat pump retrofit can be economically attractive.

The installation of tangential blowers would improve the performance of hydronic heat pump systems, but might prove impractical. For example, the placement of blowers on the old style cast iron radiators may be too obtrusive for consumer acceptance. Also, the blowers may generate drafts and levels of noise that may be uncomfortable.

Packaged terminal heat pumps offer few benefits in comparison to other retrofit options. Basically, the operating cost was inferior to the hydronic heat pump system, the oil displacement performance was poor and there could be installation difficulties. Its only advantage is the provision of summer cooling.

It was recommended that the findings of the study, which were based on laboratory tests and computer simulations, be verified through a monitored field demonstration.

WATER-SOURCE HEAT PUMPS

Institut de Recherche de l'Hydro-Québec of Varennes, Québec, completed this study on extended range water source heat pumps in March 1978.

PROJECT RATIONALE

Although water source heat pump installations date back to the thirties, their growth in popularity has been slow.

The objective of this study was to identify and analyze the factors governing installations of water-source heat pumps in Canada. These factors include:

- the availability of suitable water sources
- the design and operation of water source heat pump systems
- the treatment and disposal of water
- the environmental implications.

WATER SOURCES

Four potential sources of water were studied, well-water, lakes, rivers, and sea-water. Only well-water was found to be practical. Lakes, rivers, and seas are generally inaccessible and provide water below required temperatures.

Well-water sources were assessed with reference to the requirements of water source heat pumps available on the market, i.e. minimum 0.4 L/s flow rate and 5 C temperatures. The four areas in Canada that met these criteria were the Fraser Valley in British Columbia, the Norfolk Plains and Ottawa Valley in Ontario, Prince Edward Island, and most of the inhabited areas of Québec.

DESIGN AND OPERATING FACTORS

Three factors were identified in the design and operation of a water source heat pump system. Efficiency dictates a minimizing of start-up cycles, because a heat pump does not provide heat instantly on start-up. Suggestions for reduction included the use of a variable speed compressor and an increase in the thermal inertia of the house being heated. The second factor identified was the type of heating system used to distribute the heat. Different fluid temperatures and COP's were realized with different types of heat distribution systems, i.e. fin tubes, radiators, panel heating, and forced air. Systems using low temperature circulating fluids are preferred. The global COP was estimated to be from 3 to 3.5.

Two methods of supplying the evaporator with heat source water were discussed. The first method involves the direct application of the source water to the evaporator, either by immersing the evaporator in the source water or by pumping source water through the evaporator, as shown in Figure 1. A second method entails an intermediate heat transfer loop between the evaporator and the heat source water. This permits the evaporator to be located farther from the heat source. Source water pumped directly through the evaporator is more effective of the two.

WATER TREATMENT AND DISPOSAL

Water treatment may be required for problems associated with poor quality water - corrosion, scale formation, biological fouling and suspended fouling. It was felt that all problems could be solved but the treatment necessary could only be determined based on site-specific data.

Surface and sub-surface disposal methods were discussed. The recharge or sub-surface method has two options. A second well, required in both options, is used to dispose water to either the same aquifer or a different one. A disadvantage associated with the single aquifer recharge is the potential for cooled

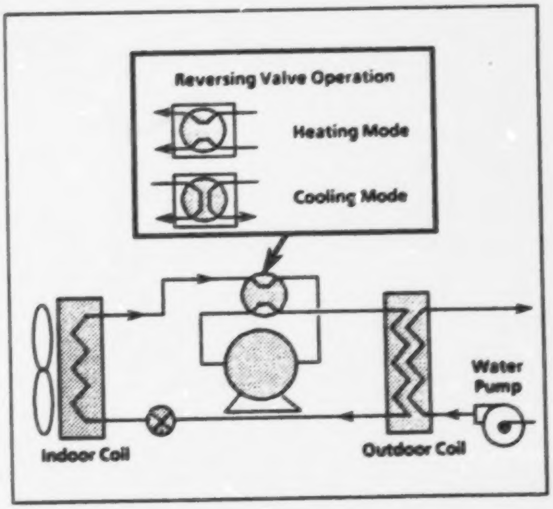


Figure 1. Water Source Heat Pump

water to migrate back to the pumping well. Disposing discharge water to a different aquifer may disturb the hydraulic equilibrium.

Surface disposal methods entail discharge to a drainage basin of gravel buried below the frost line. The effectiveness of this technique is dependent upon the permeability of the soil surrounding the basin.

ENVIRONMENTAL AND LEGAL IMPLICATIONS

Environmental problems associated with well-water source heat pump systems are thermal pollution, hydraulic imbalances in water courses, subsidence of the soil, and vegetation disturbances through careless discharge or water table level alterations.

Only one legal aspect applied. In Ontario, a permit was required to withdraw well-water if the flow rate exceeds 0.54 L/s.

CONCLUSIONS

Well water is the best source for water-source heat pumps. Regions in British Columbia, Ontario, Prince Edward Island and Québec show the most promise.

Infiltration basins provide the simplest and cheapest method of disposal.

The highest efficiency for the heat pump system is achieved when start-up cycles are minimized and an appropriate heating system is applied. A global COP of 3 to 3.5 was predicted.

RECOMMENDATIONS

Optimization and economic feasibility studies of well water heat pump systems are required.

Further studies are necessary to identify more sources of suitable water.

Design and implementation of systems for study should proceed.

WELL - WATER HEAT PUMPS

MLM Ground-Water Engineering of Edmonton performed three regional hydrogeologic feasibility studies between January 1979 and December 1981. Because of the duplication of content of the three reports, they are presented below in one project summary.

PROJECT RATIONALE

Well water offers, a high specific heat and a temperature typically 1.5 C higher than the mean annual ambient air temperature in most locations in Canada. In addition, the chemical content of well-water is not a critical factor in the operation of a heat pump and the heat energy is completely clean and renewable.

The potential for application of well-water heat pumps is dependent on two criteria: well water flow rates and temperature; and methods of discharge water disposal.

The objective of each study was to assess the applicability of well-water as a source of heat for heat pumps in three Canadian regions: the Atlantic Provinces; Ontario; and the Prairie Provinces.

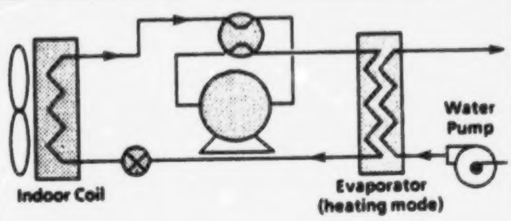
WELL-WATER REQUIREMENTS

Minimum standards for a viable well source supply were determined in relation to commercially available heat pump requirements. Information was gathered from six manufacturers of water-source heat pumps. It was established that minimum aquifer flow rates of 0.3 L/s to 0.7 L/s and temperatures of 4 C to 16 C were required for reliable operation of the heat pumps. The optimum selection was a unit requiring 4.4 C water at 0.6 L/s or 10 C at 0.4 L/s. In addition, no water quality restrictions applied, other than it be non-corrosive.

A schematic of a water - source heat pump is shown in Figure 1. Manufacturers stated COP's ranged from 2.64 to 4.5 with larger values normally associated with higher input temperatures. Installed costs are also shown in Figure 1.

DISCHARGE WATER DISPOSAL

Discharge water can be disposed of either above or below the ground surface. Above ground it can be disposed into an excavation pond, or allowed to flow into a natural drainage course. Below ground the water can be returned to the same aquifer from which it came (single aquifer method), or introduced into another aquifer (multiple aquifer method).



Input Temperature Required:	minimum 4 C
Flow Rate Required:	minimum 0.3 L/sec
Cost:	
Well Source Heat Pump	$3,400
Recharge Well (if required)	$1,000 to $2,000
Conventional Heating System	$2,000 to $2,500

Figure 1. Water Source Heat Pump

The surface disposal techniques examined were direct discharge to surface drainage courses and discharge into ponds. Direct discharge to the drainage course is simpler and cheaper but may entail flooding and freezing problems. The chief disadvantage of this method is that it may lead to depletion of the aquifer, subsidence, fracturing or saltwater intrusion. The alternative surface disposal method entails collection of the discharge in an excavation (pond) from which it is subsequently allowed to infiltrate the water table. This is shown in Figure 2. The feasibility of this method then ddepends upon the permeability of the soil and the

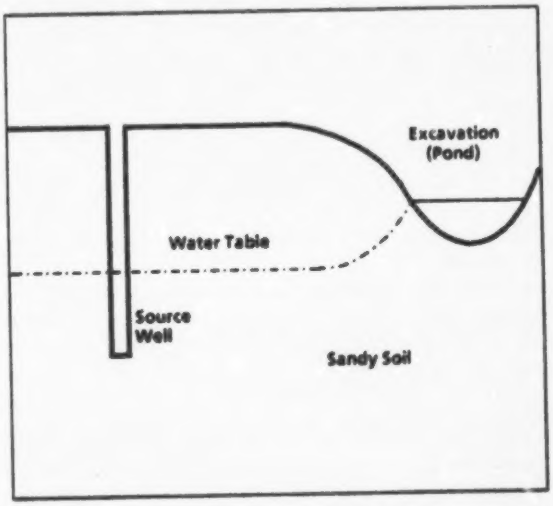


Figure 2. Surface Disposal - Excavation Discharge

depth of the water table. The excavation method is particularly effective where surface material is sandy and where recharge is designed to enter a near-surface aquifer. Because a large area is required, excavations are more practical when used collectively. An additional problem may be encountered with microbial and bacterial growth in infiltration passages.

A single aquifer sub-surface disposal method entails recharging the same aquifer supplying the heat pump. The single aquifer method was considered the best alternative because it entails no conservation or environmental deterioration in the quality or quantity of water. It also eliminates problems of saltwater intrusion, subsidence and fracturing. The main disadvantage is the possibility of thermal short circuiting, i.e. returned water migrating to the extraction well. This would lower extraction temperatures and reduce the effectiveness of the heat pump. The pumping well must be separated from the recharging well by at least 40 m to avoid thermal mixing. Three disposal well configurations are offered: a horizontal shallow well (Figure 3); a large diameter shallow well (not illustrated); and a small diameter well (Figure 4).

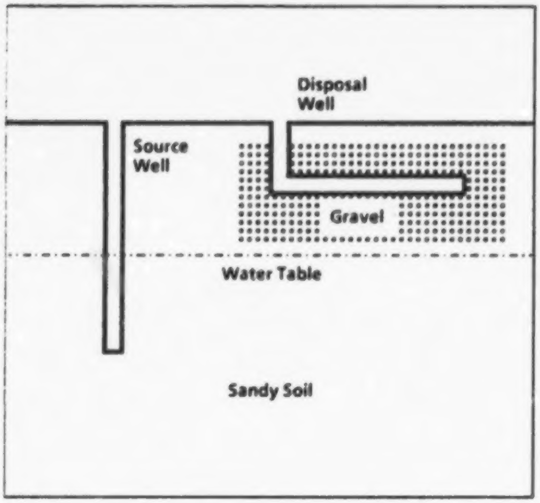

Disposal Well
Source Well
Gravel
Water Table
Sandy Soil

Figure 3. Single Aquifer Disposal-Shallow Horizontal Well Recharge

In a dual aquifer system, where a second aquifer is used for disposal, the key factor is the chemical compatibility of the water streams. If the streams are incompatible solids may precipitate causing the aquifer to clog. Alternatively, the physical matrix may dissolve, leading to the collapse of the aquifer. A dual aquifer well configuration is shown in Figure 5.

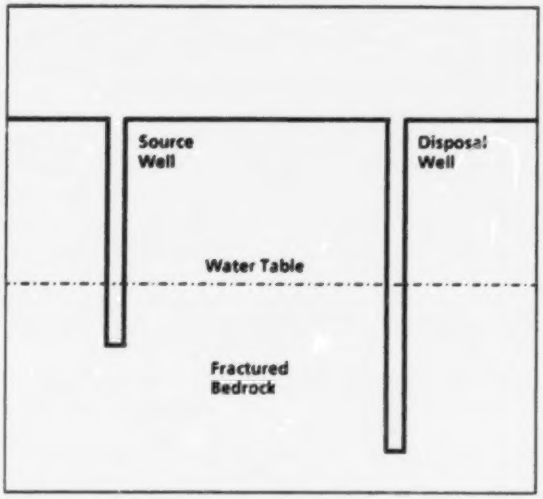

Source Well
Disposal Well
Water Table
Fractured Bedrock

Figure 4. Single Aquifer Disposal-Deep Well Recharge

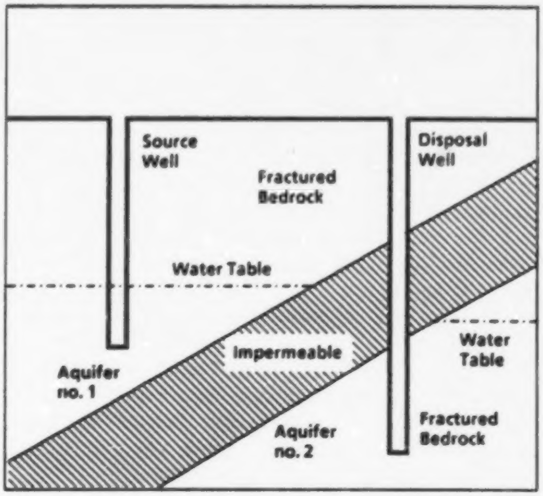

Source Well
Disposal Well
Fractured Bedrock
Water Table
Impermeable
Water Table
Aquifer no. 1
Aquifer no. 2
Fractured Bedrock

Figure 5. Dual Aquifer Disposal

WELL-WATER DATA: PRAIRIE PROVINCES

Approximately half of the area in the southern two thirds of Alberta have bedrock aquifers capable of producing greater than 0.4 L/s of water. Most of the well-water is at least 5 C.

In Saskatchewan, flows and temperatures of well-water in the areas surrounding Regina and Saskatoon appeared adequate.

In Manitoba, a large bedrock aquifer located between Lake Manitoba and Winnipeg. yields greater than 1.9 L/s with temperatures between 5 C and 7 C.

Two concerns were expressed about discharge water disposal methods. In the long term, well-water temperatures may drop because of the long heating season in the prairies despite aquifer recharge. This remains undetermined. The second concern, depletion of the aquifer when surface disposal is practiced, was felt to be unfounded providing aquifer was capable of yielding greater than 0.4 L/s and it was located in a rural area of low population density.

WELL - WATER DATA: ONTARIO

In Northern Ontario (north of 47th parallel) surficial aquifers are located sporadically throughout the Pre-Cambrian Shield region, generally yielding 0.4 L/s, but are often of small areal extent.

The most productive bedrock aquifers lie in the Hudson Bay Lowlands. Well-water temperatures at shallow depths are very low, near freezing. No data is available for deeper water from the carbonates, but it is very likely that these will be as much as 8 C warmer than those extracted from shallow depths due to geothermal heating.

In Southern Ontario (south of 47th parallel), surficial aquifers are most numerous and productive in the St. Lawrence Lowlands. Others are located in the Georgian Bay/Lake Simcoe area. These generally yield at least 0.4 L/s.

Bedrock aquifers are numerous and highly variable in yield and quantity. Well-water temperatures in Southern Ontario were predicted to range from 5 C to 10 C. Well-water yields the lowest in areas of crystalline bedrock of Pre-Cambrian age.

A small diameter vertical injection well to the source aquifer is the preferred discharge water disposal system.

Generally, hydrogeological data was much more abundant in Southern Ontario than in the Northern part, due to higher population and a corresponding increased demand for well-water.

WELL - WATER DATA: ATLANTIC PROVINCES

New Brunswick temperature data was limited to the southeast region of the province, which ranged from 7 C to 10 C. Bedrock aquifers on the east coast of New Brunswick yield from 1.9 to 7.6 L/s; those in the centre from 0.4 to 1.9 L/s; those in the northwest and along the south coast less than 0.4 L/s. Surficial aquifers, primarily located near rivers, produce from 1.9 to 7.6 L/s and greater.

Prince Edward Island contains bedrock aquifers capable of yielding 0.4 to 1.9 L/s water over the entire island but no surficial aquifers of consequence. Temperatures range from 7 C to 10 C.

Most of the well water in Nova Scotia has a temperature in the range of 7 C to 10 C. The north coast of mainland Nova Scotia and the area to the south and west of Halifax have a water temperature greater than 10 C. Most of Nova Scotia has bedrock aquifers capable of yielding from 0.4 to 1.9 L/s with several surficial aquifers with equivalent capacities located around the province.

Newfoundland bedrock aquifers are low yield (less than 0.4 L/s) except for the west coast, portions of the interior and the southeast region. Very few surficial aquifers were reported. No temperature data was available for Newfoundland.

In the Atlantic Provinces, most domestic wells lie in bedrock aquifers. These are subject to fracturing and adjacent aquifers must be considered as one hydraulic unit. The discussion of subsurface disposal methods was, therefore, restricted to the single aquifer method. Three disposal well configurations are offered: a horizontal shallow well; a large diameter shallow well; and a small diameter well. The shallow wells are not well suited to the bedrock aquifers found predominantly in the Atlantic Provinces. The small diameter well is the most economical and best suited for most Atlantic locations.

The above regional summaries of well-water availability and temperature are presented on maps attached to each of the three regional reports.

CONCLUSIONS

Accurate and extensive well water temperature data is lacking throughout the regions studied. Where surficial aquifers are of ample thickness and areal extent, they generally have adequate well-water yield. Bedrock yields are highly variable. Much of the temperature and yield information presented in the reports had to be extrapolated.

Water disposal techniques, ranked according to minimal adverse environmental impact, would be: single aquifer sub-surface recharge; double aquifer sub-surface recharge; infiltration pond; and surface drainage.

The economic attractiveness of the well-water heat pump system is enhanced when a suitable well is already available.

RECOMMENDATIONS

Well-water temperature measurements should be taken in areas where the use of heat pumps appears feasible. Well-water temperature measurements could be made by well drillers, during well development and pumping tests.

Though further research on methods to prevent temperature decline in aquifers is recommended in the reports, it would be appropriate to first study the amount of decline that will actually occur. Work in ground source heat extraction indicates an initial drop is followed by temperature stabilization.

Also, additional hydrochemical research should be done on the compatibility of the source water with the disposal aquifer where dual aquifer disposal is employed.

A set of guidelines should be prepared for each province or region, setting forth precautionary measures to guard against well-water wastage and contamination brought about by heat pump operation, especially in geologically sensitive areas, and where user population is high.

WELL - WATER SOURCE DISTRICT HEATING

Western Ground-Water Consultants Limited of Winnipeg carried out this well-water district heating study between September 1980 and March 1982.

PROJECT RATIONALE

The technical feasibility of ground-water based heat pumps for residential heating has evolved over the decade. There have been applications of both high and low temperature geothermal energy sources and, more recently, shallow well-water sources. Though few high temperature geothermal energy sources exist near major population centres in Canada, low temperature sources are available in the form of shallow aquifer systems. In Winnipeg, for example, well-water has been used as a cooling water source for commercial applications for over 30 years.

The objective of this study was to determine the technical and economic feasibility of employing either central or unitary heat pumps in a district heating scheme to supply a new 1100-home residential complex. Underground aquifers would serve as the heat source and sink.

DESIGN

The design of the district heating scheme was based on the following terms of reference:

- to supply a new 'satellite city' subdivision of approximately 1100 homes of 100 m^2 in the Winnipeg area;
- to use aquifer conditions typical of those in Manitoba; the production and recharge well system would utilize the natural energy flow within the aquifer.

The methodology employed to simulate an operating system was as follows:

- to evaluate residential heating/cooling loads;
- to estimate soil, rock and water parameters (i.e. porosity, temperatures, heat capacity, etc.);
- to analyze water flow and sizing requirements for the heat pump system and sizing of the piping network;
- to design several alternatives for the piping and recharge wells. Both piping and recharge wells were required since well-water would be used non-consumptively;
- to simulate the complete energy storage/retrieval system over several heating systems to verify that the aquifer system could be maintained at a thermal equilibrium.

RESULTS

The simulation revealed that the aquifer system would only meet the energy requirements of the

DATA:

Process:	Residential heating, cooling, DHW
Heat source:	Well water at 5 to 7 C
Heat pump	
Type:	Geo-thermal
Output:	10.6 kW
COP:	2.9
Capital cost / house:	$6,540
Annual Operating Cost (including air conditioning):	$475

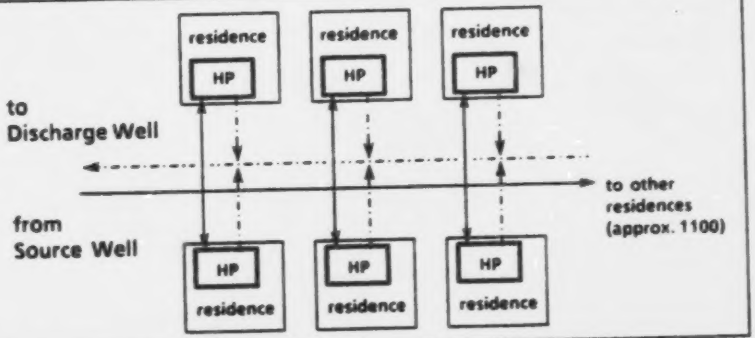


Figure 1. Well water Source Distributed Heat Pump System

In the illustration above, an aquifer is used as the energy source and sink for a new 'satellite' concept subdivision of 1100 homes. A distributed system proved more applicable than a central heat pump system. Though technically feasible with 'off-the-shelf' components, economics and aquifer recharge considerations indicate that a well-water source heat pump district heating system is currently impractical for urban areas.

proposed 1100-home subdivision for 5 to 7 years. For use as a long term energy source (approximately 90 years) a 3000 m spacing would be required between the production and recharge system.

The only practical method of coupling an aquifer system with such a district heating scheme would be to utilize the Aquifer Thermal Energy Storage concept. Here, the aquifer would serve as a storage media rather than an energy source by balancing the winter heating load with an equal amount of energy from solar energy collection systems or industrial cooling loads.

ECONOMICS

Capital costs for both the central and distributed heat pump district heating system were developed. The distributed system proved more applicable based on capital costs and practical considerations.

For the central system, the total capital cost was estimated to be $7,922,000, or $7,200 per house. This compares with a total capital cost of $7,190,920, or $6,540 per house for the distributed system. Retrofit cost would be 150% of figures quoted.

CONCLUSIONS

Well-water source heat pump district heating systems are impractical in urban areas because of the spacing required between production and recharge wells - 3000 m. Even in isolated rural areas this would present logistical problems.

The life cycle costing analysis demonstrated that there is no energy escalation scenario that would make district heating competitive with natural gas or resistance heating over the short term (15 to 30 years) or the long term (95 years). In addition, though the Aquifer Thermal Energy Storage concept would address the problem of prohibitive spacing between production and discharge wells, costs of such a scheme deem it impractical.

RECOMMENDATIONS

Renewable and waste energy sources should be evaluated to determine the most practical and cost effective Aquifer Thermal Energy Storage concept. It should be added that this course is only reasonable if significant reductions in additional costs per household are expected. Current costs range from $3,500 to $7,500.

Table 1. Life Cycle Costs / House (1981 Dollars, i = 15%, n = 30 years)

SYSTEM DESCRIPTION	CAPITAL COST(1)	ANNUAL HEATING ENERGY COST(5)	PRESENT VALUE OF HEATING ENERGY COST(6)	ANNUAL MAINTENANCE COST(7)	PRESENT VALUE OF MAINTEN-ANCE COST	PRESENT VALUE OF TOTAL COSTS
i) Distributed well-water heat pump system(2)	7,000	310	9,176 (high) 7,130 (low)	210	3,822	19,998 (high) 17,952 (low)
ii) Distributed well-water heat pump system credited with the capital cost of a central air conditioner(3)	5,500	310	9.176 (high) 7,130 (low)	165	3,003	17,844 (high) 15,633 (low)
iii) Conventional gas furnace(4)	1,000	570	16,872 (high) 13,110 (low)	30	546	18,418 (high) 14,656 (low)
iv) Electric furnace	1,000	700	20,720 (high) 16,000 (low)	20	364	22,084 (high) 17,364 (low)

(1) Cost of common ductwork not included.
(2) Includes cost of supplemental resistance heating.
(3) Credits heat pump system with installed cost of central air conditioner.
(4) Includes cost of chimney.
(5) Based on 1981, Winnipeg utility rates.
(6) It has been assumed that the cost of electricity and natural gas will escalate at the same rate. Two escalation rates have been considered:
(high) 4% above general rate of inflation (16%)
(low) 2% above general rate of inflation (14%)
(7) 1981 maintenance costs were estimated at 3% of the system capital cost for systems i, ii and iii and 2% for system iv (Includes compressor replacement for i and ii).

POWER PLANT WASTE HEAT SOURCE DISTRICT HEATING

MacLaren-Lavalin Inc., Engineers, Planners and Scientists, Toronto, completed this feasibility study in November 1981 .

PROJECT RATIONALE

The municipal water supply system, available in most communities in Canada, is a potential source of heat with a much narrower range of temperature fluctuation than the outdoor air. A water-source heat pump would have a higher COP in mid winter than an outdoor air-source heat pump.

With the new planned town of Townsend an abundant supply of heat was available at minimum cost from the condenser cooling water discharge of a large thermal plant. A municipal water treatment plant sat almost next door. The water mains were in place and the homes planned but construction barely started. This presented an excellent opportunity to determine the economics of a district heating scheme utilizing residential water-source heat pumps.

The objective of this study was to determine the technical and economic feasibility of utilized power plant waste heat and a municipal water distribution system to provide district heating.

DESIGN

The heat source is condenser cooling water drawn from the Nanticoke Thermal Generating Station on Lake Erie. This would, though an indirect heat transfer scheme, warm treated water at the Region of Haldimand-Norfolk Water Treatment Plant 2 km away. The warm treated water would then be distributed to all houses in the new townsite of Townsend, approximately 15 km away, through the municipal water distribution system. Homes in this community would use this warmed water supply as the heat source for water-source heat pump space heating.

The current capacity of the municipal water system is determined by the summer 'maximum day' demand. The winter 'maximum day' water use is less, however, leaving the difference between the summer and winter 'maximum day' consumption rates available for use by heat pumps. This capacity is sufficient to allow approximately 20 percent to 30 percent of the Townsend households to use water source heat pumps without any need to increase capacity at the water plant or in the distribution network.

In order to minimize water consumption, water would be recirculated through the evaporator until its temperature is reduced to 2 C. Drainage of discharge water can be readily handled in the storm water drainage system; discharge to the sanitary drainage system is avoided since no water treatment is required for the heat pump discharge.

Heat losses between the water treatment plant and a typical dwelling in Townsend were determined. A transport efficiency of 52 percent was calculated, with the majority of loss taking place in the short service lines feeding individual houses. Through life cycle cost calculations, the cost of insulating these lines was found to be three times as much as the resultant savings. Without insulation, the temperature of the water reaching the homes would range from 9 C to 11 C.

ECONOMICS

Various economic analyses were provided to cover differing conditions: i.e. dwelling sizes; percentage of dwellings using heat pumps; current and projected costs of resources; current and projected growth of the community and municipal water system; and heat pump sizes. It was found that the alterations to the municipal water system would cost approximately $1.1 million (1981) and could be recovered if the water charge rate was 36.8¢/m^3 (see Table 1).

Table 1. Capital Cost of Heating System

	Cost
New equipment installation and modifications on existing facilities	$1,090,000
Total Annual Cost (Amortized capital cost)	$190,000
Break-even Charge Rate for Warm Water	36.8¢/m3

The total annual cost to own and operate a 4.2 kW heat pump was $1280, assuming a water cost of 36.8¢/m3. Gas heating was found to be substantially less expensive, with an annual owning and operating cost of $759. These costs are shown in Table 2. Also shown in Table 2 is the cost of owning and operating a heat pump if the water cost is 18.3¢/m^3 (the average Ontario rate) and 13.5¢/m^3 (the cost at which the annual operating cost of a gas furnace and a heat pump are equal).

Table 2. Annual Cost Comparison
(Small Sized House, Water Use = 786 m^3/ yr)

Costs	Gas Furnace	Heat Pump		
		Water 13.5¢/m^3	Water 18.3¢/m^3	Water 36.8¢/m^3
Operating	$404	$404	$442	$586
Owning	$292	$544	$544	$544
Maintenance	$ 63	$150	$150	$150
Total	$759	$1098	$1136	$1280

In addition it was calculated that warmed municipal water would reduce total DHW costs by $46,000, or $12 per home, based on 3900 homes in the Townsend, Jarvis and Hagersville areas using the warmed municipal water supply by 1988.

CONCLUSIONS

Technically, the system proposed is feasible, with little additional plant extensions required. A water supply of approximately 10 C could be supplied to dwellings at a break-even cost of 36.8¢/m^3. This would place annual operating costs well above gas heating but below electrical resistance heating. Heating costs for DHW would be reduced by $12 per year for all homes using the municpal water supply.

The system could serve a maximum 31 percent of the dwellings of Townsend. It is expected that costs will have to be below the break-even point in the early years to encourage growth in installations.

RECOMMENDATIONS

A small demonstration program should be attempted to confirm the promise of this scheme. Initially, unheated water, at subsidized rates, could be used.

It should also be recommended that the water-source heat pumps be compared with air-source heat pumps as well as natural gas and electric heating systems. The proposed scheme might also prove more beneficial in areas of cold winter ambient where air-source heat pumps are not economical.

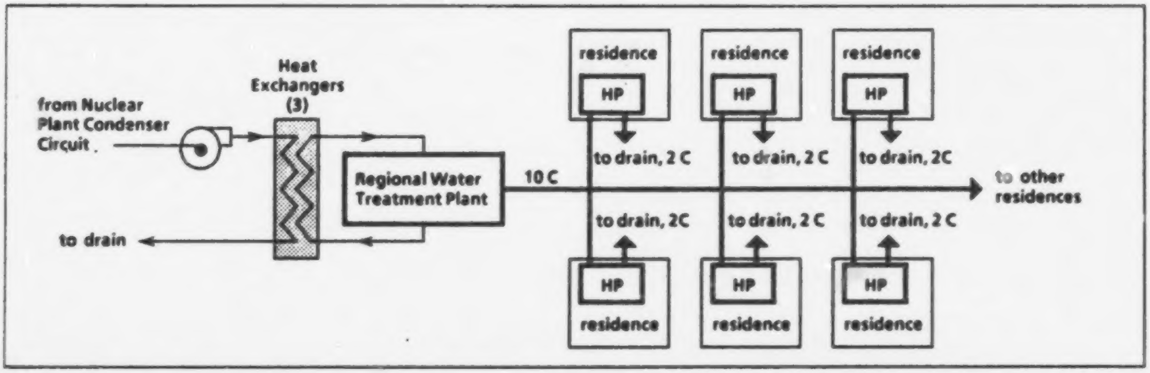


Figure 1. Nuclear Power Generating Plant Condenser Cooling Water Circuit Supplying a Distributed Heat Pump System

In the illustration above, warm water from a generator plant condenser is supplied to a water treatment plant. Using water-to-water heat exchangers, heat is added to treated water. This is then distributed to households with heat pumps. Technically feasible, the scheme entails annual operating costs per individual dwelling above those of natural gas and below electric heating.

INDUSTRIAL WASTE HEAT SOURCE DISTRICT HEATING

Boulianne, Dauphinais, Desgagné, Tremblay and Associés Inc. of St. Félicien, P.Q. carried out the following district heating feasibility study between June 1983 and July 1984.

PROJECT RATIONALE

One alternative for a district heating scheme would be to utilize a source stream from a large, constant, industrial process coupled to individual heat pumps, providing a community's heating energy requirements. Such an arrangement addresses the potential available in small settlements with a resource industry or manufacturing plant.

The objective of this project was to study the feasibility and economics of a heat pump district heating system.

OVERVIEW

The project site was a village of Ste. Méthode, Québec. The study proposed that clean industrial wastewater be piped to approximately 75 of the 81 buildings in the village, and be used as the heat source for individual water source heat pumps. The heat pumps would supply heating, cooling, and DHW. The wastewater would be an effluent stream from the Donohue St. Félicien Kraft pulp and paper plant. This stream, which is currently rejected in the Mistassini River, would be continuous, except for a short maintenance shutdown in January, and would have an average temperature of 20 C.

SYSTEM DESIGN

The proposed distribution network was divided into six strategic zones to serve the village. Zones were determined according to three factors: allowable friction losses and pressure control in the piping; distribution to buildings; and water and sewer system design. Piping from the plant to the heat pumps would utilize polyvinylchloride (PVC) which proved superior to polyethylene, cast iron, and asbestos.

With the availability of large quantities of warm wastewater available, it was determined that the heat pump should be water-to-water or water-to-air. The Command/Aire model proved most satisfactory because it was found to provide heating, cooling and DHW for approximately two-thirds of the cost of others ($290,000 vs. $450,000).

A preliminary study of a central heat pump design led to the conclusion that it was not a viable

DATA:

Process:	Residential heating, cooling, DHW
Waste heat source:	Kraft paper mill wastewater effluent
Operating hours:	Continuous
Heat pump output:	10 kW
COP:	3.3
Capital cost:	$1,084,000
Savings:	$108,000/year
Payback:	>16 years

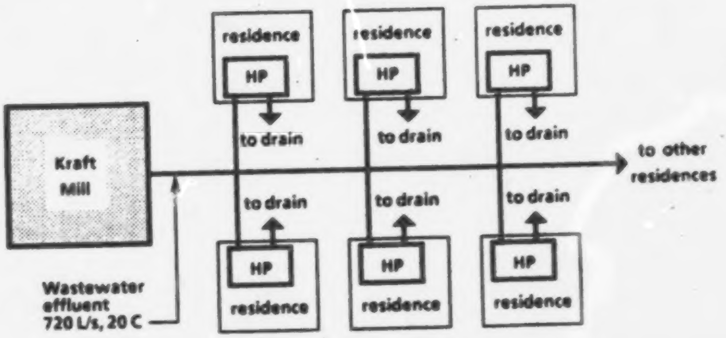


Figure 1. Application of Residential Heat Pumps to a District Heating Scheme, located in St. Methode, P.Q.

The distribution network outlined above is described in the study. Wastewater effluent from the kraft paper mill is piped to heat pumps located in buildings within the village. The heat pumps are expected to supply heating, cooling and domestic hot water to those buildings in the network.

alternative. One heat pump would have served the village, with upgraded service hot water distributed to each building; however, it would prove three times as expensive and would not be capable of supplying the DHW or air conditioning.

ECONOMICS

Total annual savings were calculated against actual consumption costs for electricity, oil or these in combination with wood. The savings were based on a heat pump COP of 3.3 and include DHW heating costs. The economics were as follows:

Capital Costs

Distribution Network	$ 794,000
Heat Pumps	$ 290,000
Total Cost	$1,084,000

Savings

Energy Savings:	$ 121,000
Less Distribution Network Operating Costs:	$ 13,000
Net Savings:	$108,000

Simple Payback	10 years
Discounted Payback	greater than 16 years

The capital costs would be returned through an annual municipal tax levied on all properties. The municipality would administer the heating systems.

CONCLUSIONS AND RECOMMENDATIONS

A system was designed for piping the wastewater stream from Donohue St. Félicien Kraft plant to heat pumps in the buildings of Ste. Méthode.

Based on a heat pump COP of 3.3 a discounted payback period greater than 16 years was predicted.

Municipalities are not empowered to operate or manage district heating systems. The provincial government, therefore, would have to provide a special ruling, allowing the town council to create a management corporation.

No recommendations were made, but a proposal was prepared by the consultant for a follow-up detailed engineering design.

PERFORMANCE OF AN AIR-SOURCE HEAT PUMP

National Research Council of Canada, Ottawa, completed the following field performance monitoring between December 1978 and May 1980.

PROJECT RATIONALE

In December 1977, an air-source heat pump was installed in one of the Mark XI energy conservation research houses in Ottawa. The system was purported to incorporate a number of features recognized as important in a northern climate heat pump.

The objective of this study was to monitor and record:

- the heating energy supplied and consumed by the air source heat pump system under normal operation (in response to the house load) and steady state conditions (artificially induced load);
- the frequency, duration and energy consumption of the defrost cycle.

HEAT PUMP SYSTEM

The heat pump system was a split system of nominal 9 kW capacity with three 4.6 kW stages of electric resistance heaters.

Features to improve the defrost cycle performance were incorporated. These included "demand" defrost initiation, wider than normal fin spacing on the outdoor coil, hot gas entry from the bottom during defrost, and a suction line accumulator on the compressor.

The compressor was designed with improved motor cooling and lubrication to enable operation under low-load conditions.

INSTRUMENTATION

Experimental measurements included:

- heat supplied by the heating system (measured by a copper-constantan thermopile across the indoor unit);
- energy used by the heating system (measured by watt transducers);
- position of reversing valve;
- outdoor dry and wet bulb temperatures;
- air temperature indoors at thermostat and before and after indoor coil.

All experimental data was scanned and recorded by an automatic data logger and magnetic tape unit. The raw data was analyzed and the following quantities calculated:

- the average power demand of the indoor section (electric resistance heaters and blower);
- average power demand of the outdoor unit (compressor motor, outdoor fan, crankcase heater and controls); and
- the average heating load.

From this data the heating demand satisfied by heat from the outdoor air was calculated. This in turn was used to calculate the seasonal performance factor.

RESULTS

A seasonal performance factor (SPF) of 1.5 was achieved in the 1978-1979 heating season (indoor blower ran continuously). The 1979-1980 winter was milder and an SPF of 1.7 was achieved (indoor blower cycled). The improved SPF in 1979-1980 was attributed to the milder weather and the use of 'auto' as opposed to 'continuous' indoor blower operation. Energy use is summarized in Figure 1.

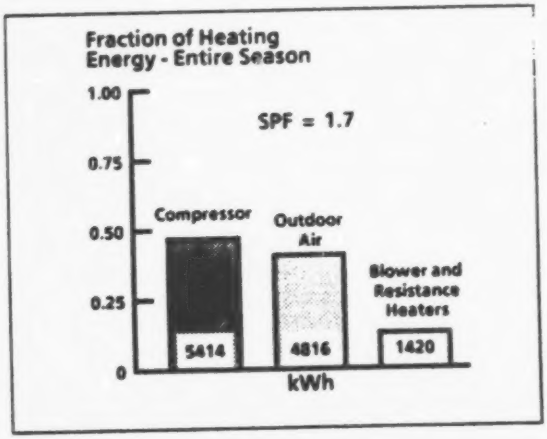


Figure 1. Heat Pump Performance Summary, 1979 - 1980

The demand defrost system averaged one cycle per day (demand defrost was initiated by high pressure drop across the air side of the outdoor coil and terminated when the refrigerant temperature leaving the coil reached 18C.) This can be compared

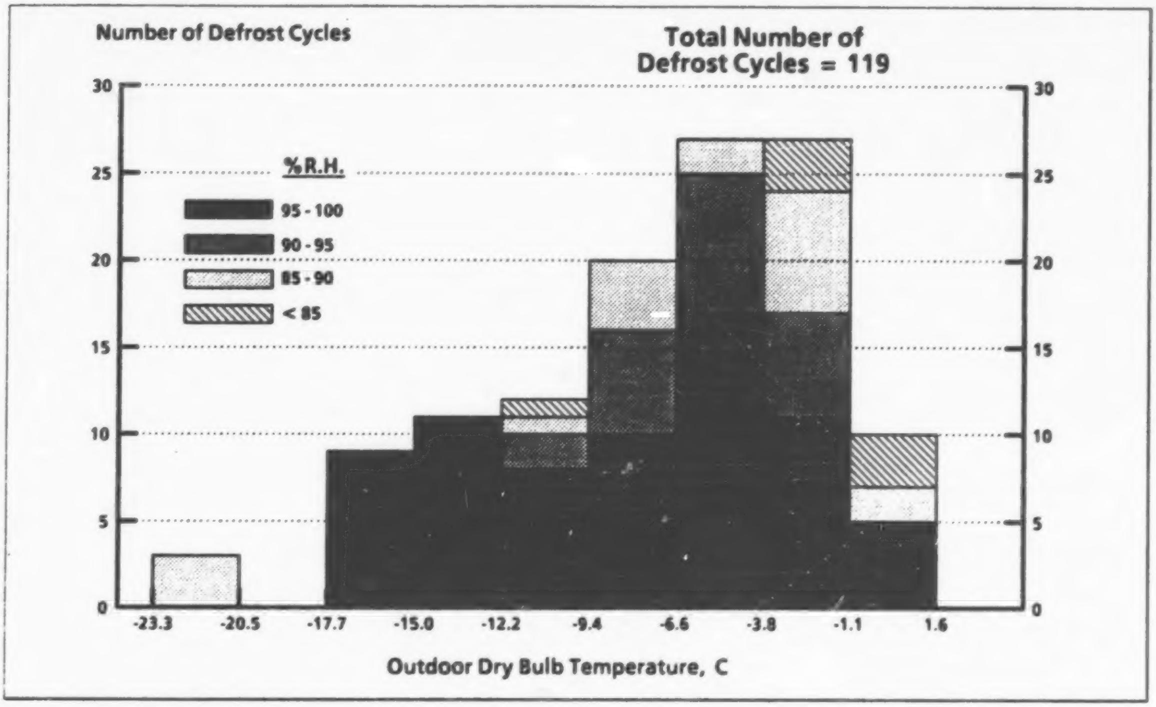


Figure 2. Number of Defrost Cycles as Function of Outdoor Dry Bulb and Relative Humidity

to an anticipated 7 cycles per day for a conventional time-temperature controlled defrost system. In terms of total defrost energy consumption, the demand system used less than 1 percent of the total energy use compared to an estimated 5 percent for a conventional time-temperature defrost system. The number of defrost cycles as a function of outdoor dry bulb temperature and relative humidity is shown in Figure 2 above.

The heat pump's 'on-off' operating performance was compared to its instantaneous performance under the same outdoor conditions. For example, instantaneous COP of 2.35 was measured compared to a COP of 2.05 under 'on-off' operation at 2 C outdoor temperature.

CONCLUSIONS

Based on the monitored results of a state-of-the-art (1977) air-source heat pump installed in a well-insulated house in Ottawa, 30 percent to 40 percent of the annual space heating load can be supplied from energy in the outdoor air.

The demand defrost system was an improvement over the conventional time-temperature defrost control method. It substantially reduced both the number of cycles and the amount of energy associated with defrost operation.

RECOMMENDATIONS

A cost benefit analysis will be required to determine the most appropriate sizing procedure to employ for Canadian climatic conditions.

EXHAUST AIR HEAT RECOVERY HEAT PUMP

The National Research Council undertook the following field performance investigation of an exhaust air heat recovery heat pump installed in the HUDAC Mark XI project, in Ottawa, in the winter of 1983-84.

PROJECT RATIONALE

In houses where natural ventilation rates are low, or moisture production rates high, the resulting high relative humidity levels can lead to moisture-related building problems. An exhaust fan can be operated to increase the ventilation rate to reduce the relative humidity level. The increased ventilation rate, however, increases the building heating load and the cost of heating. By recovering energy from the exhaust air stream, a heat pump can lower the relative humidity and the overall cost of space heating.

The objective of this project was to evaluate a prototype exhaust air heat recovery heat pump's effectiveness as a supplementary heater, its effect on house air change rate and its performance in actual use.

SYSTEM DESIGN

The prototype, designed and built by KeepRite Inc. for NRC had the following design features:

- 5 row evaporator coil with 3.9 fins per cm and 2.7 kW capacity at -20 C evaporating temperature with an inlet air flow of 52 L/s at 20 C;
- 5 row condenser with 4.72 fins per cm and a heat rejection capacity of 4.3 kW at 38 C condensing temperature with an inlet air flow of 260 L/s at 20 C;
- Tecumseh high efficiency (R22) heat pump compressor with a capacity of 7.9 kW at standard rating conditions;
- centrifugal fans for both the condenser and evaporator, the latter driven by a single speed, the former by a three speed direct drive motor;
- solid state defrost timer to initiate defrost cycles every thirty-four minutes.

INSTALLATION, OPERATION & TESTING

The exhaust air heat recovery heat pump was installed in combination with an existing electric furnace in an unoccupied house in the HUDAC Mark XI project. The house had a natural ventilation rate of 0.5 air changes per hour at an indoor-outdoor temperature difference of 35 Celsius degrees. The air exhausted from the house was drawn from the basement at a rate of 57 L/s. The air flow over the condenser was drawn from the basement, at a rate of 191 L/s and discharged into the return air plenum connected to the central electric furnace. The electric furnace fan ran continuously during the testing.

The exhaust air heat recovery heat pump's operation was controlled from the first stage of a two stage space thermostat. The electric furnace operated as required from the second stage to supplement the output of the heat pump.

Instrumentation and metering equipment were installed to measure the heat output and electrical energy use of the exhaust air heat recovery heat pump and the electric furnace.

The following tests were undertaken during the monitoring period:

- the exhaust air heat recovery heat pump system and electric furnace were alternately used during selected periods to obtain comparative energy use;
- the air change rate of the house was measured during exhaust air heat recovery heat pump system and electric furnace operating periods to determine the impact of exhaust air heat recovery heat pump operation on the natural ventilation rate.

TEST RESULTS

The purchased heating energy required to maintain the same space temperature was less with the exhaust air heat recovery heat pump/furnace combination than with the electric furnace only. The actual energy use reduction was a function of indoor-outdoor temperature difference; 23 percent and 33 percent for temperature differentials of 40 and 10 Celsius degrees respectively.

The house air change rate and the infiltration heating load during exhaust air heat recovery heat pump operation were typically 30 percent higher than during electric furnace operating periods.

The annual purchased heating energy reduction, based on test house results, was estimated to be approximately 5000 kWh. This represents a 30 percent energy saving in comparison to the same house without an exhaust air heat recovery heat pump.

The system operated without problems. The defrost cycle worked well and the operating noise of the unit blended in well with normal heating system sound levels.

CONCLUSIONS AND RECOMMENDATIONS

The exhaust air heat recovery heat pump was effective in supplying a significant portion of the heating energy in the test house, while increasing the base infiltration or air change rate by approximately 30 percent.

The exhaust air heat recovery heat pump's impact on relative humidity levels was not assessed in this project. Since humidity control would be a prime motivating factor in purchase of such a unit, a performance investigation in an occupied home is warranted.

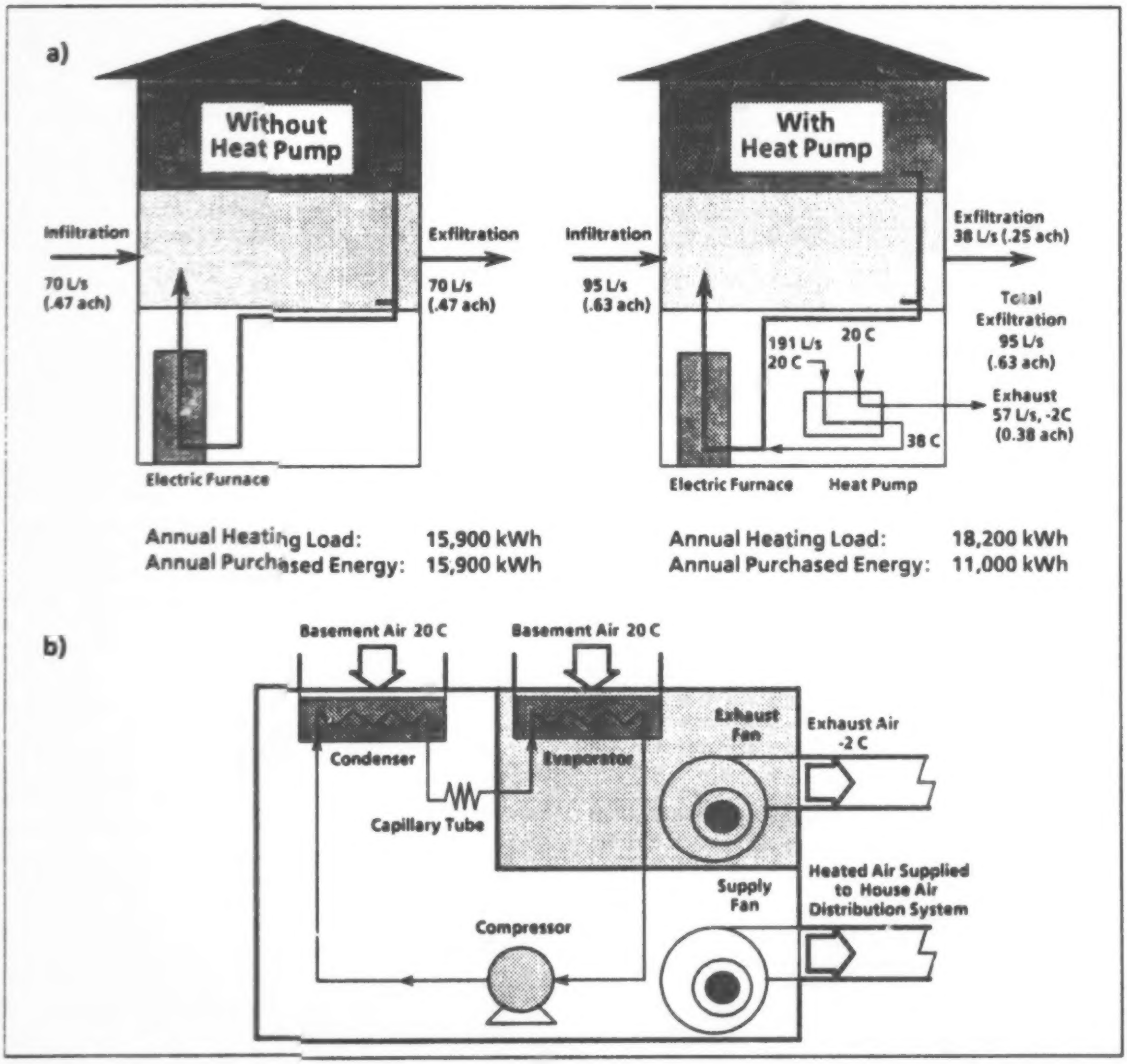

a)
Without Heat Pump
Infiltration
70 L/s (.47 ach)
Exfiltration
70 L/s (.47 ach)
Electric Furnace
Annual Heating Load: 15,900 kWh
Annual Purchased Energy: 15,900 kWh
With Heat Pump
Infiltration
95 L/s (.63 ach)
Exfiltration 38 L/s (.25 ach)
Total Exfiltration 95 L/s (.63 ach)
191 L/s 20 C
20 C
38 C
Exhaust 57 L/s, -2C (0.38 ach)
Electric Furnace
Heat Pump
Annual Heating Load: 18,200 kWh
Annual Purchased Energy: 11,000 kWh
b)
Basement Air 20 C
Basement Air 20 C
Condenser
Evaporator
Exhaust Fan
Exhaust Air -2 C
Capillary Tube
Supply Fan
Heated Air Supplied to House Air Distribution System
Compressor

Figure 1a) Effects of Exhaust Air Heat Recovery Heat Pump on Infiltration Rates and Annual Energy Consumption

Figure 1b) Schematic of the Exhaust Air Heat Recovery Heat Pump

HYBRID HEAT PUMP/ FOSSIL FUEL FURNACE

Further to previous development work for the National Research Council of Canada, this project was carried out by Chromalox Canada Inc., and completed in October 1982.

PROJECT RATIONALE

An add-on heat pump's seasonal heating efficiency could be improved if allowed to operate continuously below the balance point. There are currently no units commercially available which are capable of such operation.

A combined fossil fuel furnace / heat pump package, or 'hybrid' heat pump, could operate in this manner, but would have to be designed to overcome two major obstacles. The first is the problem of furnace heat exchanger corrosion. For continuous operation below the balance point, the indoor coil must be located upstream from the furnace heat exchanger. This configuration, however, can lead to furnace heat exchanger corrosion due to condensate formation during the cooling season. The second obstacle is the differing air flow requirements of a furnace and a heat pump.

The objectives of this project were to:

- design and build several hybrid heat pumps;
- evaluate their performance through laboratory and field trials;
- assess their market potential.

DESIGN

Two sizes of hybrid heat pumps were designed: a 7 kW and an 8.8 kW unit, both of which incorporated an indoor coil located in the return air duct upstream from the furnace heat exchanger. Approximately one-half the air flow through the indoor coil is diverted to the furnace heat exchanger. The remaining air flow passes through a bypass duct with fixed damper, located in parallel with the furnace heat exchanger. This was incorporated into the design to satisfy the difference in air flow requirements between the furnace and the heat pump. The furnace heat exchanger was constructed from corrosion resistant aluminized steel and conformed to existing regulations for upstream cooling coil location. Spark ignition rather than a standing pilot was employed. A schematic of the hybrid heat pump is shown below in Figure 1.

Commercially available components were used in the construction of the prototypes. The design objectives were heating capacities of 8.8 and 7 kW with a COP of 3.0 at 8.3 C.

PERFORMANCE EVALUATION

The laboratory testing was carried out in dual psychrometric rooms in accordance with ASHRAE standards. Heating and cooling capacities and efficiencies were determined for each size of hybrid heat pump. The corrosion resistant furnace heat exchanger was subjected to 1500 hours (equivalent of six cooling seasons) of operation in the cooling cycle, and results showed no sign of internal corrosion.

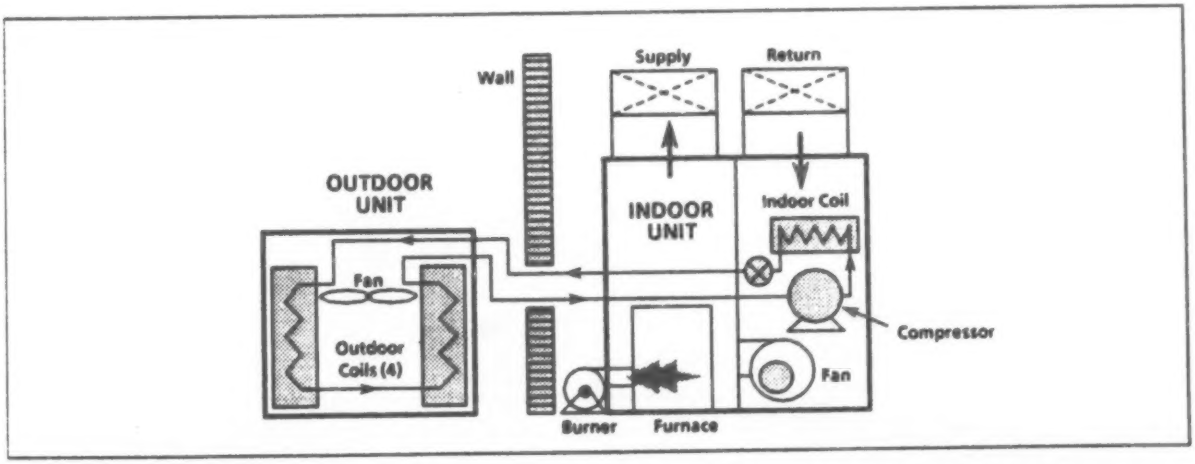


Figure 1. 'Hybrid' Heat Pump - simultaneous heating below the Balance Point shown

Field testing and monitoring was carried out in four homes of similar size and construction, two being equipped with gas 'hybrid' heating systems and two with oil 'hybrid' systems. Field performance monitoring included the power consumed by the heat pumps, thermocouple temperature readings and furnace air flow measurements. Gas was also used for service water heating in the two homes equipped with gas furnaces. The water heating load was estimated from summer billings, rather than measured by a separate meter.

Several factors would tend to distort the test results. The insulation was upgraded in one of the oil-heated houses during the period of field testing, and the new occupants kept the temperature lower and practiced night setback. In addition, the new hybrid system furnaces installed in the test homes were downsized from their predecessors, and had higher steady-state efficiencies.

The results of the laboratory and field tests indicate that more than 60 percent of the fossil fuel used by a conventional furnace can be displaced by the hybrid heat pump. The field test results are presented in Table 1.

MARKET POTENTIAL

It was determined that the oil furnace / heat pump combination offered an attractive payback. Such is not the case, however, for the gas furnace / heat pump combination. It was estimated that the consumer installed cost for a 7 kW hybrid system would be approximately $5000, a capital cost 9% higher than that of a new furnace with a conventional add-on heat pump.

CONCLUSIONS

It was estimated that several basic design deficiencies lowered the seasonal efficiencies of the prototype units by 10 to 12%, and that additional design modifications could result in a further 25% improvement in seasonal efficiencies. These modifications would include:

- a more suitable high efficiency compressor selection;
- coils matched to the compressor, with a more effective fin design;
- modifications in refrigerant circuitry and refrigerant expansion technique;
- an alternate method for defrost control, such as 'outdoor motor load' demand defrost;
- a more efficient outdoor coil air flow design;
- a direct-drive furnace fan with a high efficiency motor;
- an inexpensive oil stripping heat exchanger.

Evaporator flooding occurred during the field testing due to the horizontal placement of the outdoor coils. For this reason, any future design would employ a vertical coil arrangement. A demand defrost control system which senses pressure drop across the coil rather than outdoor motor load was used, due to the unavailability of a control system for the latter.

It was determined that the oil furnace / heat pump combination offers an attractive payback; however short term production volumes foreseen at the time of the study were insufficient to encourage tooling up by the manufacturer.

Table 1. Results of Hybrid Heat Pump Field Testing

Test House	Heat Pump Size (kW)	Reduction in Fuel Use		Heat Pump Electricity Use (kwh)	Heating Cost Savings* ($)
		Gas (m^3)	Oil (L)		
A	8.8	1660 (62%)	-	6229	100
B	7.0	1260 (63%)	-	5304	50
C	8.8	-	3020 (77%)	6660	630
D	7.0	-	2070 (73%)	7130	317

* 4.5¢/kWh, 23¢/m^3, 30.8¢/L

SPACE HEATING HEAT PUMP WITH DHW PREHEAT

W. L. Wardrop and Associates Ltd. carried out monitoring of this experimental system from March 1981 to March 1982, and completed their report in December, 1982.

PROJECT RATIONALE

Domestic hot water heat exchangers which are presently available for heat pumps use only refrigerant superheat to preheat domestic hot water and operate only when the heat pump is supplying heating or cooling.

In a previous study, a new design was developed which provided water heating not only from refrigerant superheat, but also from heat of condensation, and liquid subcooling. Water heating was possible both on space heating and cooling cycles, and on an additional 'water heating only' cycle.

The objective of this project was to modify a commercially available air source heat pump to preheat domestic hot water, as outlined above and to evaluate it's performance in the field.

DESIGN

The system evaluated, shown below in Figure 1, was a two-speed air-to-air Lennox heat pump, supplemented by an electric furnace. A water-cooled condenser (WCC) and a liquid subcooling heat exchanger (LSHE) were added to the heat pump to provide water heating. The WCC was shop fabricated, as an appropriate heat exchanger was not available at the time. The WCC recovered refrigerant vapour superheat and the heat of condensation, depending on the operating mode, while the LSHE recovered heat from liquid refrigerant. Water preheated by these two heat exchangers is stored in two preheat tanks: circulated through the WCC during heat pump operation, and drawn off through the electric hot water storage tank as required. Supplementary water heating was provided, as required, by resistance heaters.

The control system gave priority to the space heating load, with excess heat pump capacity applied to water heating. High-speed compressor operation was used from -25 C to 0 C, while low-speed operation was used from -10 C to 25 C. The high-speed balance point was -10 C: the low-speed balance point 0 C. Water heating was locked out below the high-speed balance point temperature.

PERFORMANCE EVALUATION

The system was installed in a Winnipeg home, and monitored daily for a period of one year. The heat pump system was operated over a wide range of outdoor temperatures. It was determined from performance data to have met 52 percent of the water heating load and 48 percent of the space heating load during the year of monitoring. Water

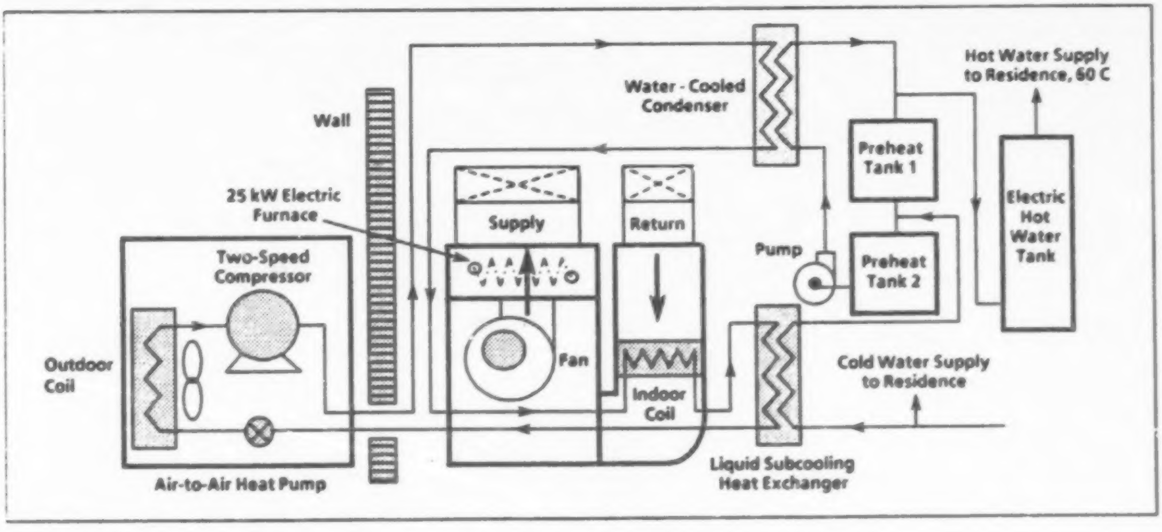

Hot Water Supply to Residence, 60 C
Water - Cooled Condenser
Wall
Preheat Tank 1
25 kW Electric Furnace
Supply
Return
Electric Hot Water Tank
Two-Speed Compressor
Pump
Preheat Tank 2
Outdoor Coil
Fan
Cold Water Supply to Residence
Indoor Coil
Liquid Subcooling Heat Exchanger
Air-to-Air Heat Pump

Figure 1. Water Heating Heat Pump System - Heating Mode Shown

heating costs were reduced by 15 percent when compared with electric resistance heating.

System breakdowns and incomplete data readings, coupled with the fact that the winter, during the monitoring period, was one of the coldest recently experienced in Winnipeg, resulted in much lower water heating savings than had originally been anticipated.

The summer during which the system was monitored was unseasonably cool, providing little opportunity for water heating on the space cooling cycle. The heat pump had sufficient cooling capacity to meet the cooling load on low speed, therefore the high-speed cooling cycle was locked out. Although little water heating was obtained on the space cooling cycle, the system performed well at outdoor temperatures up to 30 C.

Problems included the replacement of a faulty reversing valve, resulting in the shutdown of the water-heating-only cycle for one month. Also, a check valve that was found to be leaking refrigerant was replaced with a new unit on three separate occasions.

CONCLUSIONS

Several improvements to the experimental system were recommended. The compressor, suction accumulator and unit controls, which were located in the outdoor unit, should be combined with the indoor components as an indoor packaged unit. This arrangement would allow recovery of heat normally lost from these components, and they would be protected from severe cold weather conditions.

Projections made in the report estimate further energy savings could have been achieved if the following improvements to the system were made:

- the conventional preheat water storage tank and the electric hot water tank both suffered large standby losses; increased insulation would reduce these losses. In addition, the size of the electric hot water tank could be reduced by 50 percent in order to further reduce losses;

- the replacement of the air-cooled circulator pump with a water-cooled type would result in normally wasted motor heat being recovered by the water heating system;

- a smaller crankcase heater, which would activate only when the heat pump is off, would lower power consumption considerably.

Based on this 'improved' system, paybacks of 5.9 years and 8.4 years could result for water heating heat pump installations located in Vancouver and Winnipeg, respectively. The projected performance figures upon which the above paybacks depend are based on manufacturers steady-state data and average year weather information; allowances for cyclic and defrost losses are not included in the estimates.

RECOMMENDATIONS

A number of recommendations were made on features to be incorporated in future prototype designs. Laboratory monitoring of the prototype's steady state efficiency should then be followed by field installation and subsequently by additional field performance monitoring. This packaged water heating heat pump should then be reassessed in terms of economic viability.

WELL-WATER HEAT PUMP: AQUIFER CAPACITY AND SYSTEM PERFORMANCE

National Research Council of Canada, Ottawa, documented this field test in a report dated December, 1984.

PROJECT RATIONALE

A prototype of a well-water heat pump had been designed and developed by Westinghouse for NRCC. In order to assess its performance, a home in the Ottawa area was selected for a field trial. The objective of this study was to determine the heating performance of the prototype well-water heat pump.

SYSTEM DESIGN

The heat pump had a nominal capacity of 9 kW with 15 kW auxiliary electric resistance heating. The heat pump provided heating only, removing heat from a well-water supply and delivering it to the indoor coil (condenser), see Figure 1. Air conditioning is also provided, but not by the heat pump. During summer operation, well water is diverted to a separate circuit within the indoor coil. Air passing through the indoor coil is cooled by the well water.

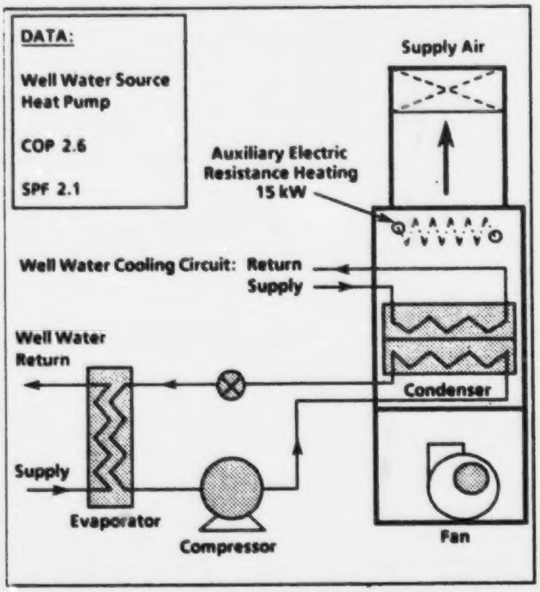

DATA:
Well Water Source Heat Pump
COP 2.6
SPF 2.1
Supply Air
Auxiliary Electric Resistance Heating 15 kW
Well Water Cooling Circuit: Return
Supply
Well Water Return
Condenser
Supply
Evaporator
Compressor
Fan

Figure 1. Well-Water Heat Pump and Well-Water Air Conditioning

Well water was supplied to the heat pump from an existing 55 m well. A new 75 m well was drilled for disposal of the water circulated through the heat pump. By returning the well water to the aquifer, depletion of the aquifer would not occur. The source and discharge wells are shown in Figure 2.

At the outset of the project the existing well was tested and the withdrawal rate of 0.4 L/s was determined. This maximum withdrawal rate approximately equaled the minimum flow rate specified for the heat pump prototype. thus establishing that this was a marginal application for a well-water heat pump. The recharge well was

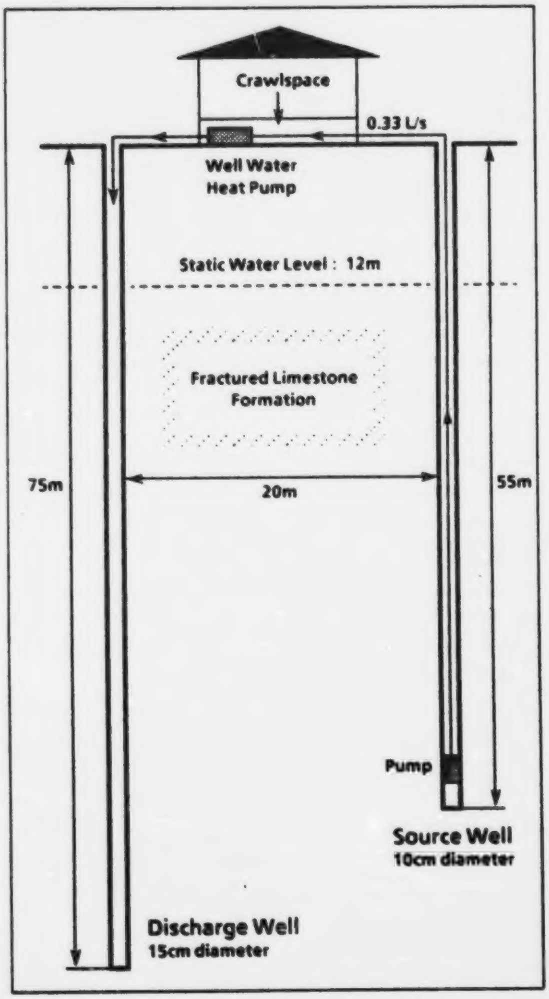

Crawlspace
0.33 L/s
Well Water Heat Pump
Static Water Level : 12m
Fractured Limestone Formation
75m
20m
55m
Pump
Source Well
10cm diameter
Discharge Well
15cm diameter

Figure 2. Well-Water Source Residential Heat Pump

tested for recharging and found satisfactory. A coincidental rising of the water level in the source well at the time of the recharge testing indicated a horizontal cross linking through the 20 metres of fractured limestone separating the wells.

TEST RESULTS

The heat pump system was installed in the fall of 1982 and was monitored from January 1983 through June 1984. Measurements included water level, temperature and flow rate, and energy consumption.

The water level in both wells, with the heat pump off, varied with the local water table, that is, rising during prolonged wet periods and dropping during extended dry periods. The average static water level was 9 m from the surface, occasionally rising to a high of 5 m and dropping to a low of 12 m. The operation of the heat pump, with a flow rate of 0.33 L/s caused the water level in the source well to drop 2 to 3 m from the static level and rise 2 to 3 m in the recharge well. The water level in both wells quickly recovered to normal static levels when the heat pump shut off. The temperature of the water pumped from the well ranged from 8.5 C in the winter to 10 C in the summer.

A summary of the purchased power to run the system and the quantity of water and energy removed from the aquifer are shown in Table 1. The seasonal performance factor, which is a measure of system efficiency, averaged 1.9 during the first heating season. Note that heat pump system data for the second heating season was not available until the end of January because of a failed water pump. A new pump was installed and proved to be 40 percent more efficient in pumping water. This resulted in an increase from 1.9 to 2.1 in the seasonal performance factor .

CONCLUSIONS AND RECOMMENDATIONS

The test results indicate that the present method of testing well withdrawal rates is conservative. The well used in this experiment when tested in the conventional manner indicated the maximum withdrawal rate was marginal for the requirements of the heat pump. After two winters of operation, no adverse reduction in available water or water temperature was observed. One additional fact which was observed was the difference in efficiency of two water pumps of the same capacity, but from different manufacturers, and their effect on the seasonal performance factor of the system.

Air conditioning performance was outside the scope of the study and was not reported. However, the heat pump performs air conditioning by circulating 10 C well water through the indoor coil. A recommendation not made in the report would be to evaluate the performance of this method of air conditioning.

UPDATE

The prototype heat pump described in this report has since been commercialized. It is licenced by the Government of Canada to Rocton Energy Systems Limited of Nepean, Ontario.

TABLE 1: Summary of the Purchased Power to Run the System and the Quantity of Water and Energy Removed from the Aquifer

Date	Quantity of Water Flowed (m^3)	Energy Removal from Water (kW) A	Power for Water Pump (kW) B	Power for Heat Pump and Fan (kW) C	Power for Resistance Heaters (kW) D	COP		Seasonal Performance Factors (kW) $\frac{A+C+D}{B+C+D}$
						Incl. Water Pump $\frac{A+C}{B+C}$	Excl. Water Pump $\frac{A+C}{C}$	
Jan./83	684.7	2990	526	1812	299	2.05	2.65	1.93
Feb./83	494.0	2090	393	1337	157	1.98	2.56	1.90
Mar./83	463.4	2050	367	1281	236	2.02	2.60	1.89
Apr./83	335.8	1490	264	949	149	2.01	2.57	1.90
May /83	163.1	700	135	461	133	1.95	2.52	1.78
Feb./84	419.6	2050	193	1267	177	2.27	2.62	2.13
Mar./84	496.0	2380	236	1457	219	2.27	2.63	2.12
Apr./84	238.0	1140	113	732	151	2.22	2.56	2.03
May /84	169.9	850	94	540	114	2.19	2.57	2.01

WELL-WATER AND GROUND-SOURCE HEAT PUMP FIELD PERFORMANCE MONITORING

Seven contracts were awarded in the fall of 1982 for monitoring the in-service performance of six well-water heat pumps and one ground source heat pump in homes in different regions of Canada. The National Research Council of Canada summarized the studies of 7 consultants in this report dated April 1985.

PROJECT RATIONALE

Water source and ground source heat pumps were increasing in popularity in 1982, which prompted the National Research Council to assess the performance and reliability of some early installations.

The objective was to determine the seasonal performance factor or annual system coefficient of performance for heat pumps extracting energy from well-water or a ground coupled heat exchanger.

MONITORING PROGRAM

The seven installations are described in Table 1. A number of problems were encountered at the Winnipeg location and monitoring was terminated before data was obtained.

Site measurements consisted of the following:

- flow rate of well water or brine (in ground coupled system in Nobleton, Ontario);
- heat transfer between the evaporator and the water or brine circuit;
- the electrical energy consumed by the compressor, indoor blower, submersible well pump or circulating pump (in ground coupled system).

The energy extracted from the water/brine by the heat pump systems was measured using energy meters; supply and return water temperatures and water/brine flow rate were simultaneously measured to compute the energy transfer rate and this quantity was integrated with time to obtain the total energy extracted. The electrical energy consumption of the heat pump systems was recorded by kilowatt-hour meters.

The measured quantities were recorded weekly and monthly performance summaries were submitted to the National Research Council.

Table 1. Installation Characteristics

Location	Residence Heat loss kW (ΔT° C)	Heat Pump		Supply Well			Water Disposal Method	DHW Supplied	COP Claimed by Mfr.
		System Type	Nominal Size (kW)	Depth (m)	Flow Rate (L/s)	Water Temp. (°C)			
Black Point Nova Scotia	12 (42)	Reversible Water source	17.6	55	0.7	9.5	Recharge Well	Portion of DHW	3.9
Fredericton New Brunswick	10 (44)	Non-reversing Water source	14.6	38	0.5	7.5	Stream	Yes	3.2
Ste. Marthe Québec	12 (50)	Reversible Water source	12	27	0.6	9	Creek	No	3.5
Nobleton Ontario	19 (39)	Non-reversing Earth source	14	N/A	N/A	N/A	N/A	Yes	2.8
Leamington Ontario	17 (37)	Reversible Water source	20	15	1.0	12	Recharge Well	No	3.3
Langley B.C.	20 (40)	Reversible Water source	13.5	10	0.75	12	Recharge Well	No	3.1
Winnipeg Manitoba	17 (53)	Reversible Water source	14.6	36	0.75	5.5	Recharge Well	No	3.8

MONITORING RESULTS

The seasonal performance factor (SPF) was defined as the total heating energy supplied (Q_{hp}) divided by the total electrical energy used by the heat pump system (P_{hp}) over the entire heating season.

The total heating energy supplied was the sum of the electrical energy used by the compressor (P_c), the electrical energy used by the indoor blower (P_b), the resistance heater electrical energy use (P_a) and the total heat extracted from the well water or brine (Q_e).

The total electrical energy use consisted of all electrical energy used by the compressor, indoor blower and resistance heaters plus the submersible well pump (P_p).

The measured performance of the six installations over the monitoring period is summarized in Table 2.

INTERPRETATION OF RESULTS

The test results showed that the SPF's for the systems as measured were significantly lower (typically 30 to 50 percent) than the coefficient of performance claimed by the heat pump manufacturers. As no auxiliary heating energy was required by five of the systems monitored (the only difference, by definition, between SPF and COP), the large discrepancy was explained as consisting of one or both of the following:

- pumping energy, which ranged from 13 to 32 percent of the total electrical energy use, is not included in manufacturer's COP ratings;
- heat pump utilization factors or percent operating times were low, which resulted in considerable on-off cycling and increased energy use due to part-load inefficiencies. Manufacturer's COP ratings on the other hand are for steady-state operation.

CONCLUSIONS

Six of the heat pump systems operated satisfactorily with SPF's ranging between 1.5 and 2.4. These values are, however, considerably below the manufacturer's performance ratings.

The SPF's of the well water systems could be improved by reducing the energy requirements for water pumping.

Table 2. Monitored Energy Use

Location	Monitored Period	Compressor Energy (kWh) P_c	Indoor Blower Energy (kWh) P_b	Water/ Brine Pump Energy (kWh) P_p	Auxiliary Heating (kWh) P_a	Total Purchased Heat Pump Energy (kWh) $P_{hp} = P_c + P_b + P_p + P_a$	Energy Recovered from Water/ Brine (kWh) Q_e	Heating Energy Supplied by H.P. to house (KWh) $Q_{hp} = P_c + P_b + P_a + Q_e$	SPF $\frac{Q_{hp}}{P_{hp}}$	COP Claimed by Mfr.
Black Point Nova Scotia	02/02/83-01/25/84	10917	1770	2354	—	15041	16347	29034	1.9	3.9
Fredericton N.B.	09/13/82-05/31/83	5514	537	1786	—	7837	12295	18346	2.3	3.2
Ste. Marthe Quebec	11/01/82-10/19/83	4636	695	1273	—	6604	10679	16010	2.4	3.5
Nobleton Ontario	11/16/782-121/17/83	14114	1108	1874	493	17589	18833	36422	2.1*	2.8
Leamington Ontario	11/08/82 11/27/83	12510	4120	2200	—	18830	25779	42409	2.3	3.3
Langley B.C.	02/24/83-02/23/84	5480	1077	814	—	7371	4786	11343	1.5	3.1

*SPF calculation includes circulating pump energy in heat supplied, because pump was located in the house.

RELIABILITY OF ADD-ON HEAT PUMP INSTALLATIONS

The National Research Council of Canada released this study in February 1985 based on information gathered during a two year monitoring period (1981-1983).

PROJECT RATIONALE

With the introduction of the Canada Oil Substitution Program Grants, the add-on heat pump became a more attractive off-oil option leading to increased sales of heat pumps in many regions of Canada. Add-on air-source heat pumps were installed in a number of occupied homes in Orleans, Ontario. A field monitoring investigation was undertaken to assess the performance of these heat pumps.

This report documents one aspect of this performance assessment, namely, the observed installation-related problems and equipment reliability of the add-on heat pumps monitored.

MONITORING

During the winter of 1981-82, seventeen heat pumps added on to natural draft gas furnaces were monitored. The following year saw one installation withdrawn from the study, while six homes with new installations were added to the study.

Prior to the monitoring period, pertinent information relating to each system was documented, including: system indoor air flow, furnace steady-state efficiency, control settings, construction details and household occupancy factors.

Two site visits, to each installation, were made each week to record submetered energy use and to observe and document system operation under varying weather conditions.

INSTALLATION-RELATED PROBLEMS AND RELIABILITY

Component failures observed over the two-year period, included:

- 3 compressor replacements (in one installation)
- one outdoor fan motor
- a solid-state control board
- two furnace gas valves
- an outdoor fan blade

Problems requiring corrective action were categorized as follow (frequency of occurrence in brackets):

- installation related (3)
  (location of outdoor unit, control wiring etc..)
- design related (13)
  (component failure, factory wiring, weather etc..)
- owner education (5)
- furnace (1)

The repeated compressor replacements in the one installation were explained as due to defective replacement compressors. No problems were experienced in the winter of 1982-83.

The fan motor failure and the cracked fan blade were due to an 'open-top' fan discharge outdoor unit design located under a roof dripline. Ice accumulations under certain weather conditions eventually led to the failure.

In the owner education category, complaints of low air temperatures discharging from supply grilles was the most frequently cited problem. Large volumes of low temperature supply air, at outdoor conditions at or below the heat pump balance point, was the reason. Homeowners had not been forewarned of this inherent characteristic of heat pump operation.

Table 1 summarizes the corrective actions (including component failures) taken over the two year period. In total, there were 22 corrective actions taken by the installing contractor or NRCC technicians over this time. This represents 0.54 corrective actions per unit-year of operation.

CONCLUSIONS

The corrective actions taken per unit year of operation (0.54) was significantly lower than that reported by Ontario Hydro in a previous study (1.33). Installation and component reliability had improved significantly over the intervening few years.

More attention needs to be paid, by the designer, to the configuration of the outdoor unit to avoid weather-related operational difficulties.

The industry needs to explain, in simple terms, the operational characteristics of add-on heat pumps to homeowners to ensure that they obtain the maximum benefit from the system

Table 1. Corrective Actions During 1981-82 and 1982-83 Heating Seasons

Date	House No.	Problem	Corrective Action	Type of Problem	Corrective Action By
Oct.8, 1981	11	Unit not operating	Reset High Pressure switch; Power off for 24 hours; Unit off on High Pressure when power restored	Owner education	NRCC
Oct. 23, 1981	19	Fan blade iced-frozen-unable to turn	Removed ice blockage	Installation (location)	NRCC
Nov. 13, 1981	8	Furnace gas valve burnt -control wiring	Replace gas valve -remove factory jumper between thermostat power terminals	Installation (control wiring)	Installing contractor and NRCC
Dec. 3, 1981	10	Control wiring -furnace appears to have provided most of heating through fall	Disconnect wire -not used in add-on application	Installation (control wiring)	Installing contractor
Dec. 10, 1981	11	Outdoor fan blades hitting ice buildup	Removed ice buildup	Design (weather related)	Installing contractor
Dec. 10, 1981	17	Cold air discharge at grilles	None	Owner education	Installing contractor
Dec. 21, 1981	7	Compressor prevented from running by defective printed circuit board	Replace solid-state control board	Design (component failure)	Installing contractor
Dec. 21, 1981	13	Compressor burnout (defective start relay)	Replace compressor (installed Jan.29/82)	Design (component failure)	Installing contractor
Jan. 4, 1982	8	Service light on -control wiring	Owner turned off system - notified contractor-problem in factory wiring	Design (factory wiring of interface control)	Installing contractor
Jan. 4, 1982	17	Cold air discharge at grilles	Contractor told owner to switch to emergency heat -NRCC told owner to leave heat pump on	Owner education	Installing contractor
Feb. 5, 1982	1	Gas pilot valve failure	Gas pilot valve replaced	Furnace component failure	Ottawa Gas
Mar. 1,1982	7	Outdoor fan motor failure -bearing failure	Replace outdoor fan motor	Design (component failure)	Installing contractor
Mar. 15, 1982	13	Replacement compressor had defective relief valve	Replace compressor	Design (component failure)	Installing contractor
Mar. 22, 1982	13	Replacement compressor could not be started on bench	Order new compressor	Design (defective component)	Installing contractor
Mar. 30, 1982	19	New owner complains about low discharge air temperature at grilles	Explain that 'low' temperatures are normal	Owner education	NRCC
Nov.15, 1982	8	Compressor off on high pressure	Reset high pressure switch - not certain of cause (defective plenum thermostat replaced later)	Design (defective component)	NRCC
Nov. 19, 1982	4	O.D. fan blade cracked	Owner switches to emergency heat - orders new blade	Design (defective component)	Owner
Jan. 12, 1983	20	Low discharge air temperature at grille	Owner instructed contractor to cut resistor on circuit board to prevent heat pump operation below - 12 C	Owner education	Installing contractor
Feb. 4, 1983	15 20 25	Discharge grille on outdoor units covered with ice following freezing rain storm	Remove ice	Design (weather related)	NRCC
Feb.14, 1983	25	Owner dissatisfied with system performance (high gas usage)	Install control to allow operation below balance point	Design	NRCC and Installing contractor

PRESSURE REGULATED HEAT PUMP (R22)

The National Research Council of Canada produced the following laboratory report concerning residential heat pumps, dated September 1981.

PROJECT RATIONALE

A heat pump suitable for use in northern climates requires a system with low air conditioning capacity, and a high heating capacity at low outdoor temperatures. To achieve this, industry has tried a number of design approaches. These, however, have all introduced complexities into an already complex machine, and new technology with new problems and high first cost to the consumer. The need exists for a simpler, more cost effective adaptation.

The objective of this study was to develop an air source heat pump design suitable for Canadian climatic conditions.

OVERVIEW

Heating needs were met by designing a heat pump system with a compressor that was 30% larger than required for the cooling load. This effectively lowered the balance point by 5 Celsius degrees and decreased the hours of wintertime auxiliary heating by 20 to 30 percent. During summer operations, compressor suction is restricted by the crankcase regulator to maintain efficient operation.

SYSTEM DESIGN

Two modified systems were designed - one to compare with a 7 kW standard heat pump and the other to compare with an 8.8 kW heat pump. Both utilized crankcase pressure regulators and larger compressors. Some of the advantages of the resulting system were:

- larger system heat exchangers because maximum must-run tests were no longer a restriction;
- less power requirement for the compressor motor due to restriction of high load conditions;
- smaller duct losses due to lower indoor air volumes;
- utilization of standard off-the-shelf components;
- high overall efficiency.

Two disadvantages were given for the regulated heat pump system. First, condensation would gather on components downstream of the regulator, requiring extra insulation and corrosion protection. Also, because of reduced cooling efficiency, a 1 percent increase in annual cost of energy would be realized during 800 air conditioning hours.

RESULTS

Modified heat pumps were compared with equivalent conventional systems installed in identical houses in two geographical locations, Toronto and Ottawa, Ontario. These locations were chosen because their summer design conditions are nearly equal but they have a larger difference in heating degree days and winter design conditions. The COP for the 6 kW and 8.8 kW systems were estimated for Toronto and Ottawa locations, as shown in Table 1.

Table 1: Comparison of COP for Standard and Regulated Heat Pumps

	7 kW System		8.8 kW System	
	Toronto	Ottawa	Toronto	Ottawa
Standard	2.32	2.23	2.27	2.18
Regulated	2.62	2.56	2.47	2.41

For the 7 kW system, this resulted in 32 percent less supplementary heat required in Toronto and 25 percent less in Ottawa, while for the 8.8 kW system this resulted in 46 percent less supplementary heat required in Toronto and 35 percent less in Ottawa.

CONCLUSIONS

The results of this analysis support a design approach for northern heat pumps by compressor suction pressure control. The regulator reduces the complexity of the system and all parts are standard state-of-the-art devices.

DATA

Heating capacity:	30% higher than cooling capacity
Regulator setting:	-1 C
COP (7 kW):	2.56 and 2.62
(8.8 kW):	2.41 and 2.47
Supplementary heat reduction:	25% and 32%

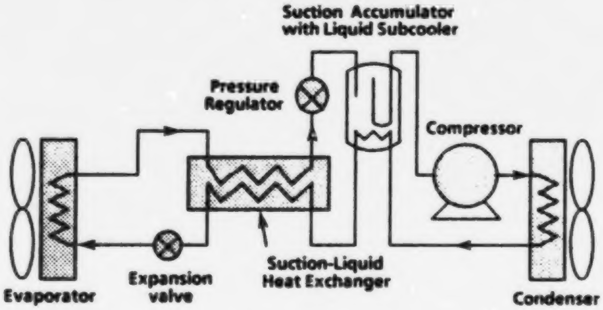

Suction Accumulator with Liquid Subcooler
Pressure Regulator
Compressor
Evaporator
Expansion valve
Suction-Liquid Heat Exchanger
Condenser

Figure 1. Northern Climate Heat Pump with Pressure Regulator (for simplification only the heating mode is shown).

In the illustration above, a crankcase regulator has been added to a 'standard' heat pump design. In summer conditions the regulator controls the downstream pressure to a maximum of the regulator setting. This, in effect, reduces the system capacity. Compressor efficiency is thus maintained in a system which was 30% larger than the cooling load required. In the winter the system operates at full capacity to meet the heating needs.

Operations, otherwise, remain the same in the regulated system. In winter, refrigerant at the 'evaporator' recovers heat from the outside air, becomes superheated by the liquid-suction heat exchanger, is compressed at the compressor, then rejects its heat at the condenser inside the building. In summer, heat exchanger functions are reversed. Heat is now absorbed at the indoor heat exchanger (providing cooling to the building) and rejected to the environment at the outdoor heat exchanger.

PRESSURE REGULATED HEAT PUMP (R502)

The National Research Council of Canada produced the following laboratory report concerning residential heat pumps, dated October 1981.

PROJECT RATIONALE

Residential heat pumps have traditionally been designed for milder climatic conditions than those found in Canada. The emphasis has been placed on the heat pump's cooling capacity, not on its heating capacity. Overall efficiency should be improved as heat pump performance is modified to better match ambient conditions.

The objective was to compare the operation of a pressure regulated heat pump using R502 refrigerant with a conventional heat pump using R22, for use in northern climates.

OVERVIEW

A refrigerant/compressor system with medium and low temperature efficiency was necessary to meet Canadian climatic conditions. R502 was chosen because it had the lowest boiling point (-45.6 C) at atmospheric pressure of any of the most used refrigerants. This low boiling point would permit the heat pump system to operate with refrigerant pressures above atmospheric pressure in most northern locations. These higher pressures at low temperature operation would improve the cooling of the compressor motor and reduce the discharge gas temperatures because there would be a greater mass flow rate of gas to carry away the heat of compression. Compression ratios would also be lower. Drawbacks to the system included the facts that the R502 compressor offered few capacity increments and had limited use for full capacity in the high temperature range.

RESULTS

In another NRC report (see page 3-3), a standard residential R22 heat pump was compared with a pressure regulated version. These comparisons were made for buildings in Toronto and Ottawa, cities chosen because they have similar summer conditions but have very different winter conditions. The same buildings were used for predicting performance of the R502 regulated heat pump in this study, and the following tables compare the findings a standard R22 heat pump, a regulated R22 heat pump, and a regulated R502 heat pump.

Table 1: EER of the Air Conditioning Function

	7 kW System	8.8 kW System
Std. R22	8.1	7.5
Reg. R22	7.4	7.3
Reg. R502	6.9	5.9

Table 2: Seasonal COP On Heating

	7 kW System		8.8 kW System	
	Toronto	Ottawa	Toronto	Ottawa
Std. R22	1.7	1.5	1.5	1.4
Reg. R22	2.0	1.7	1.9	1.7
Reg. R502	2.0	1.8	1.9	1.8

A minor improvement in the performance of the R502 system over the regulated R22 system on the heating cycle was predicted, while the reverse was true in the air conditioning cycle. The net result was that the R502 regulated system is almost identical to the R22 regulated system in overall performance.

CONCLUSION

The R502 refrigerant/compressor heat pump system has some mechanical advantages over standard heat pumps, when used in northern climates. The increased costs, due to larger refrigerant piping and high R502 refrigerant prices, make it uneconomical. Improved compressor efficiency would compensate for these increased costs and make it economically feasible.

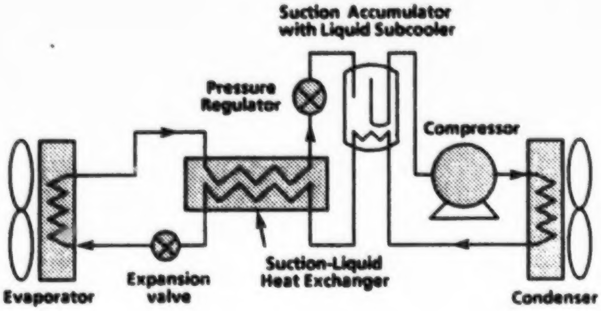

Suction Accumulator
with Liquid Subcooler
Pressure
Regulator
Compressor
Evaporator
Expansion
valve
Suction-Liquid
Heat Exchanger
Condenser

Figure 1. Northern Climate R502 Heat Pump with Pressure Regulator (for simplification only the heating mode is shown).

WATER-SOURCE HEAT PUMP

Westinghouse Canada Inc. of Hamilton, Ontario carried out this research and development project throughout 1980 - 81.

PROJECT RATIONALE

Water-souce heat pumps normally use the same water supply as a heat source (when operating in the heating mode) and a heat sink (when operating in the cooling mode). However, if cool enough water is available during the summer, the water can be used for direct cooling, eliminating the need for heat pump air conditioning.

The objective of this project was to develop and test a water-source heat pump design which would provide water cooled air conditioning.

DESIGN AND PROTOTYPE DEVELOPMENT

A number of water-source heat pump prototype units were constructed for laboratory testing and evaluation. The heat pumps had a 10.5 kW design heating capacity when supplied with water at 7 C. Standby heating was supplied by 15 kW electric resistance heaters.

The heat pumps were designed for non-reversing operation; cooling was not provided by the heat pump. Summer cooling was provided by passing cool water through a seperate circuit in the indoor coil. The cooling circuit had a design capacity of 7 kW when supplied with 7 C water. The unit is shown schematically in Figure 1.

A major design innovation of this unit was the interlacing of the heating and cooling circuits into one indoor coil, which improved the operating efficiency and reduced the material content by 20 percent. The unit was intended to be a packaged heat pump suitable for installation as a free-standing unit or added to an existing forced air heating system..

The heat pump safety controls included freeze protection for the water-to-refrigerant evaporator and high and low pressure limit switches for compressor protection. The water flow regulating controls, not shown in Figure 1, consisted of a suction-pressure regulated valve operated during the heating mode and a solenoid valve operated during the coolong mode.

Testing was carried out in the laboratory on two prototype units in order to refine the design. These tests were:

- preliminary trials to establish heat pump design and refrigerant charge;
- heat pump performance tests to obtain data for plotting performance and pressure graphs;
- blower, motor and heater tests to select components and prove safe operation. These tests included testing under abnormal operating conditions;
- refrigerant charge trials to improve heat pump

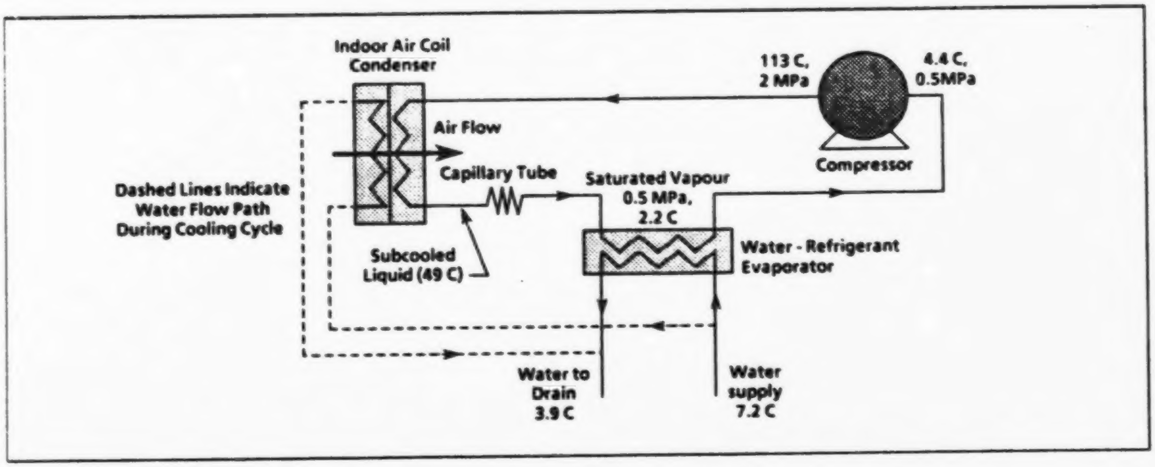


Figure 1. Water-Source Heat Pump, Heating Cycle Shown.

performance and eliminate superheat problems.

Once the design was refined, heat pump performance tests were undertaken to obtain consistent data for plotting performance and pressure graphs.

Four additional units were constructed as field test models. These were split-system versions of the prototype, with the double circuit indoor coils built into duct frames for installation in supply air ducts.

Development of this heat pump was not completed, due to the sale by Westinghouse Canada of the air conditioning division in July 1981. Evaporator freeze protection testing had only just commenced at the time of suspension of the work.

PERFORMANCE EVALUATION

Calorimeter tests indicated the water-source heat pump could be operated in the heating cycle through an incoming water temperature range of 4 C to 27 C. Actual heating capacity of the units was close to design, and the discharge and suction conditions were acceptable. It was noted, however, that a better water regulating valve may be necessary in order to increase the flow rate at low water temperature. The direct cooling capability was not evaluated because of the abrupt termination of the study.

CONCLUSIONS AND RECOMMENDATIONS

The evaporator will freeze rapidly if water flow stops too soon after the freeze-protection device is tripped. Consideration should be given to using a hot refrigerant bypass system to provide heat to the evaporator after the freeze-protection device is tripped.

It was recommended that further development and testing be carried out in order to evaluate operation of the water-source heat pump below 7 C incoming water temperature. In addition, operation of the unit in the cooling mode should be evaluated by testing.

UPDATE

A demonstration project was subsequently undertaken (page 3-33).

PACKAGED TERMINAL HEAT PUMP

Chromalox Research and Development of Mississauga, Ontario began development of a package terminal heat pump design in September 1982 and successfully concluded the project in June 1983.

PROJECT RATIONALE

Annual heat pump installations were expected to double between 1981 and 1984, primarily with add-on and all-electric heat pump sales. These heat pumps lend themselves to homes with central forced air heating systems. However, approximately 15% of Canadian homes do not have central forced air systems (for example, homes with electric or hydronic baseboard heaters); packaged terminal heat pumps could be suitable for these homes. The market for packaged terminal heat pumps would also include commercial installations, such as in condominiums, hotels and motels.

The objectives of this study were to:

- design and fabricate several prototype packaged terminal heat pumps; and
- evaluate the performance of these units, with and without enhanced heat exchangers.

DESIGN

The heat pump units were developed in sizes of 2.0 kW, 2.6 kW, 3.5 kW, and 4.4 kW; employing both non-treated and treated heat exchangers for each design. Treatment of the heat exchangers involved expansion of the tubing by using a 'bullet' expander, which produced grooving on the inner tube surfaces. This technique increased the heat transfer surface area and hence, the capacity of the heat pump.

It was the designers' intent to use only commercially available components. Maximization of Canadian content, quality of construction, ease of serviceability, and aesthetic considerations (low noise levels in particular) were all taken into consideration. Product development was carried out in conformance with CSA specifications.

One of the design goals was to obtain high COP's in the heating cycle. Computer aided design techniques were employed in balancing of the refrigerant and air-side components of the unit. The compressor selected had the highest efficiency of those commercially available. To keep the power requirements to a minimum, the heat exchangers chosen had large face areas, with low air flow face velocities. The centrifugal fan and motors were selected for high efficiency and quiet operation.

Another design consideration was that condensate disposal should not damage the building or require

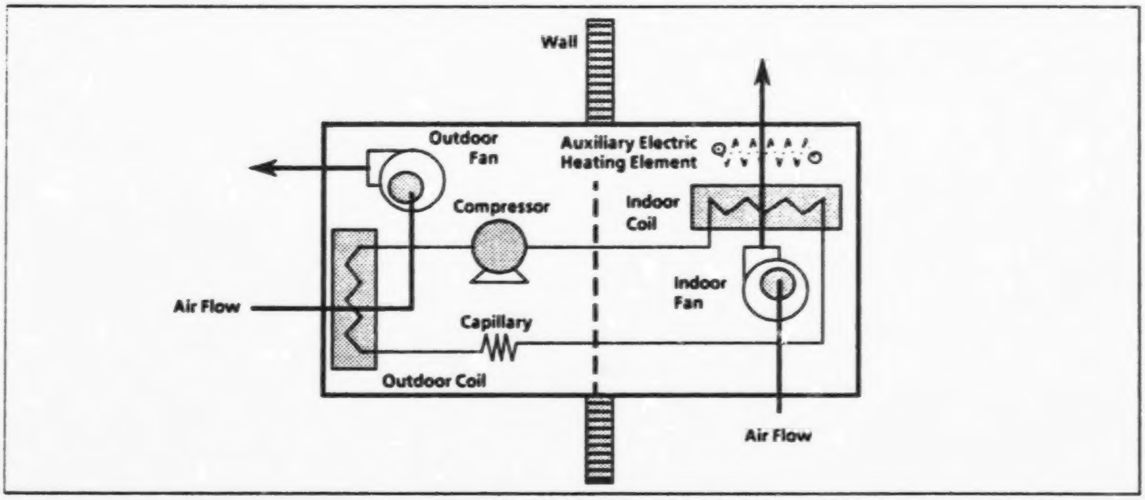


Figure 1. Air-to Air Package Terminal Heat Pump (through-the-wall installation)

drainage to a central location. Consequently a method was employed by which condensate is expelled through the outdoor fan.

Auxiliary and emergency heating is provided by electric heating elements of 1 to 5 kW, depending on heat pump size. The auxiliary heating mode is modulated by the heat pump electronic controller, while the 'Emergency Heat' circuit is turned ON / OFF at the room thermostat.

The heat pump has automatic shut-off when ambient temperatures fall below 3 C in order to avoid frosting problems.

PERFORMANCE EVALUATION

The testing of all package terminal heat pumps was carried out in accordance with Air-Conditioning and Refrigeration Institute (ARI) practice, using an in-house psychrometric calorimeter. Modifications were made to the prototype units as required during testing to 'fine tune' the theoretical design and achieve the best possible performance. One such modification was made to the centrifugal fans, ie. changing pulley sizes, to meet the desired air flow requirements over the heat exchanger coils.

The testing yielded information on the heating and cooling capacities, as well as heating cycle COP's and cooling cycle EER's. The performance testing results are presented below in Table 1, and shows the increase in capacity for each unit with and without the treated heat exchangers.

CONCLUSIONS

The effectiveness of the method used to prevent interference of the unit's operation by snow could not be evaluated, as snow conditions could not be duplicated in the psychrometric chambers.

The project resulted in a high quality heat pump which features quiet operation, high efficiency and a long life expectancy.

UPDATE

The units have been commercialized, with production commencing in December 1984.

Table 1. Air-to-Air Packaged Terminal Heat Pump Laboratory Test Results

Nominal Heat Pump Size (kW)	Non - treated Heat Exchangers				Treated Heat Exchangers				Heating / Cooling Capacity Increase (%)
	Heating Capacity (kW)	Cooling Capacity (kW)	Heating Efficiency COP	Cooling Efficiency EER	Heating Capacity (kW)	Cooling Capacity (kW)	Heating Efficiency COP	Cooling Efficiency EER	
2.0	2.31	2.20	2.10	6.4	-	-	-	-	-
2.6	2.72	2.75	2.37	7.2	2.80	2.87	2.33	7.8	3 / 4
3.5	3.45	3.44	2.46	7.6	3.53	3.55	2.72	8.4	3 / 3
4.4	4.60	4.40	2.79	8.1	4.66	4.41	2.78	8.0	1 / 0

VAPOUR COMPRESSION HEAT PUMP COMPUTER DESIGN AID

The National Research Council prepared this computer design aid, dated September 1985.

PROJECT RATIONALE

A graphically based design method has been used successfully for many years to estimate the system performance of various vapour compression (reverse Rankine) cycle systems such as heat pumps and air conditioners. The method involves the generation of compressor and heat exchanger performance curves based on manufacturers' data. It is a time consuming task when the graphs are calculated and plotted by hand, taking at least 16 hours of effort. Desk-top computers and graphics plotters now afford the opportunity to greatly reduce the time required to generate the graphs used to evaluate system designs.

The objectives of this work were to:

- describe the graphically based design method;
- develop computer software for the design method.

DESIGN METHOD

The method is essentially graphical in nature and can be done by hand or with the use of a small desktop computer and graphics plotter. Since its development in the late 1950's, it has been used to design over 20 systems that have been successfully put into production. Air-conditioning, heat pump and refrigeration units can be designed using this technique.

A vapour compression cycle's steady-state performance is essentially a result of the interaction between the evaporator, compressor and condenser. Other hardware items such as connecting lines, expansion valve, suction-to-liquid refrigerant heat exchanger have an effect, but the magnitude of this effect is usually small when compared with the influence of the above three principal components in a vapour compression cycle system. The method aids in sizing these three main components and provides an initial estimate of the system performance.

An air-to-air heat pump can be used to illustrate the method. Air flow parameters must be chosen for the condenser coil. Industry rules-of-thumb are usually used, such as 50 L/s per kW and 2.3 m/s face velocity. With these parameters and manufacturer's data, constant temperature curves are drawn for a range of discrete entering-air dry-bulb temperatures (see Figure 1a).

The condenser must reject the heat absorbed by the evaporator plus the power required by the compressor, minus losses. Experience indicates that losses can be estimated fairly well by a factor of five percent of compressor capacity. Therefore, the condenser capacity (Q_{cond}) is equal to 95 percent of the evaporator capacity (Q_{evap}) plus the compressor power input (P_{comp}) or:

$$Q_{cond} = 0.95\, Q_{evap} + P_{comp}$$

Since manufacturer's data gives compressor capacity (i.e. evaporator capacity) and power input as a function of evaporator temperature (T_{evap}) and condenser temperature (T_{cond}), discrete constant evaporator-temperature curves can be plotted as a function of condenser capacity and condenser temperature. These are shown graphically in Figure 1a, superimposed on the entering-air temperature curves.

A second graph is produced, plotting evaporator capacity (i.e. 95 percent of compressor capacity) versus condenser temperature, using compressor manufacturer's data (see Figure 1b).

For a specified evaporator, air flow rate and face velocity are estimated. Using manufacturer's data, curves are drawn (see Figure 1c) for a range of discrete entering-air enthalpies, as a function of evaporator (Q_{evap}) capacity and evaporator temperature (T_{evap}).

With a condenser entering-air dry-bulb temperature of, say 20 C, Figure 1a shows that for each evaporating temperature there will be a unique condenser capacity and condenser temperature. For example, for a -5 C evaporator temperature there will be a 10 kW condenser capacity and a 52 C condenser temperature. Each evaporator temperature (defined by the 20 C entering-air temperature) is projected vertically down to the point of intersection with its corresponding evaporator temperature curve in Figure 1b. Joining these intersections defines a curve which describes the evaporator capacity as a function of of the condenser temperature. In the example cited above, the evaporator capacity is 6.3 kW.

To determine the operating point, each of the above intersections on Figure 1b is projected horizontally across to Figure 1c until it meets its corresponding evaporating temperature projected up from the abscissa. Joining these intersections then provides a curve which describes the operation of the evaporator for the compressor, condenser and condenser entering-air temperature designated. The intersection of this curve with the enthalpy of the air entering the evaporator (33 kJ/kg in the example cited) then determines the operating point of the evaporator and hence the operating point of the whole system. Figure 1 shows the interrelationship of these three graphs. These graphs can also be used for preparing capacity and COP versus ambient air temperature curves for a particular design.

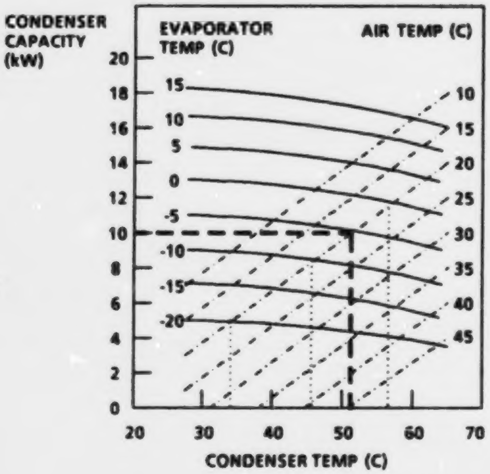


Figure 1a.

To date, the major drawback with this method was that it took considerable time to prepare the three graphs. With heat pumps the work was nearly doubled as another three graphs had to be prepared for the reverse cycle. Computer programs, written in Basic for a Hewlett-Packard computer, have been prepared which will produce the graphs of Figure 1. The programs are presented in the report.

CONCLUSIONS

Apart from its simplicity, the attractiveness of this method is that it is based solely on manufacturer's data for the compressor and heat exchangers. Therefore, it does not rely on the application of basic theory to a complex real world situation or the use of correlations based on data that does not properly describe the situation of interest. It is this feature which gives the method its high credibility as a design aid.

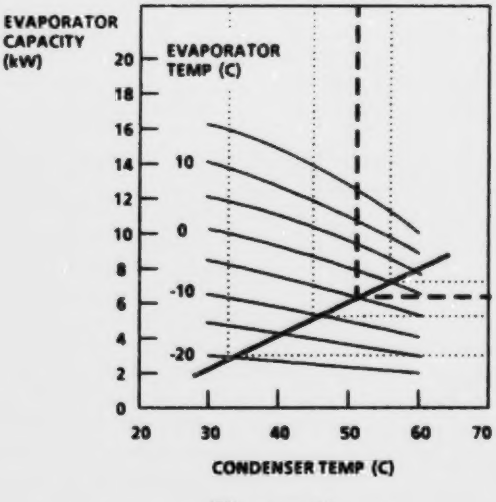


Figure 1b.

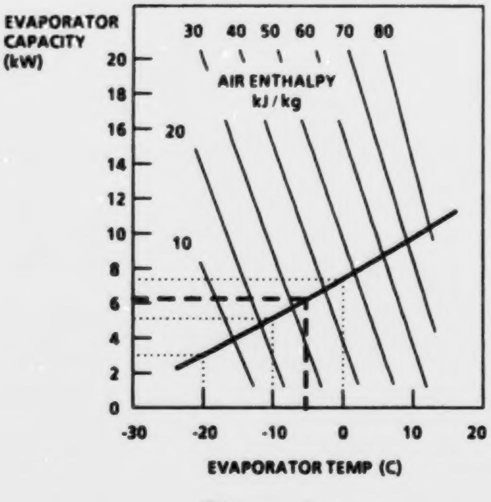


Figure 1c.

Figure 1. Example Illustrating the Use of the Vapour Compression Cycle System Design Method

PSYCHROMETRIC CALORIMETER FACILITY

The National Research Council of Canada initiated work on this heat pump test facility in March 1981 and completed construction in early 1985.

PROJECT RATIONALE

The Heat Pump Test Facility was built to provide accurate calorimetric test measurements on heat pump and air conditioning equipment under controlled psychrometric conditions. The indoor and outdoor environmental chambers are capable of reproducing a wide variation of climatic test conditions.

FACILITY LAYOUT

The facility consists basically of two psychrometric calorimeter chambers located side by side for reproducing the required combination of indoor and outdoor environmental test conditions. The computer which interfaces with the data acquisition unit and the control panels containing switching equipment, controllers, instrument displays and data acquisition system are located in an adjacent control room (not shown). The floor plan of the test facility is shown in Figure 1.

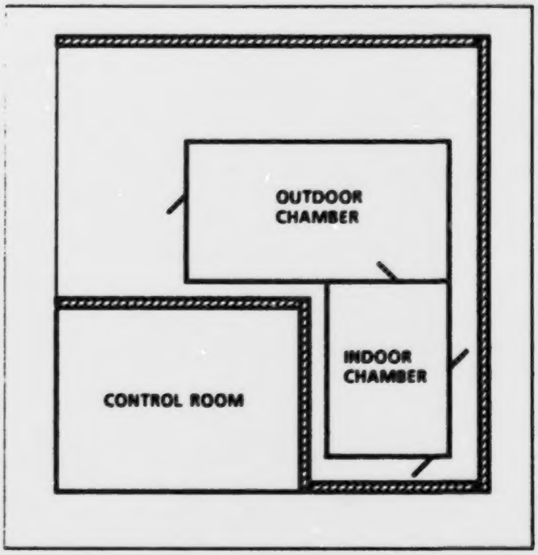


Figure 1. Floor Plan of Heat Pump Test Area.

For testing of a typical heat pump the compressor, expansion valve, outdoor coil and fan, and any auxiliary components are packaged into an instrumental test unit located in the outdoor chamber. The indoor coil is arranged within a separate instrumented test duct located inside the indoor chamber.

CALORIMETER DESCRIPTION

For each chamber the air flow is delivered by a supply fan operating in a recirculating closed circuit system. The air is continuously preconditioned by passing through cooling coils, humidifiers, and reheat coils located in ducts above the chambers.

The air flow to each chamber is cooled by passing over a cooling coil. R-11 brine pumped through the coil extracts heat which is rejected in turn by the R-502 refrigeration system. The outdoor chamber has a second cooling coil connected in a parallel duct such that the flow can be switched from one to the other in the event of a coil freeze-up. The air outlet temperature from each coil is controlled by modulating brine flow to the coil.

The original humidification system used electronic electrode steam generators to inject the required steam through distribution headers into the air ducts. However, the fixed ± 10 percent variation of steam output about the set point proved to be too large, and a simpler system was adopted using a control valve connected to a steam service line to supply the distributors. Steam injection is actuated by an input signal to a remote controller from a dewpoint hygrometer located in each chamber.

The required dry bulb temperature of air entering the chambers is maintained by a bank of electric heaters located in the air flow. The heaters are arranged in a combination of manually switched and variable blocks of power. The variable heater energy inputs are modulated by remote controllers.

Data gathered are processed by a Hewlett-Packard Model 9845C Desktop Computer. A computer program written in HP Basic language enables the operator to trigger the volt meter and take single or timed scans of all readings as required. The program converts all the voltage readings into appropriate units of temperature, resistance, pressure, flow, etc., and performs all calculations necessary. The corresponding heat pump performance results and required system parameters are tabulated and printed out.

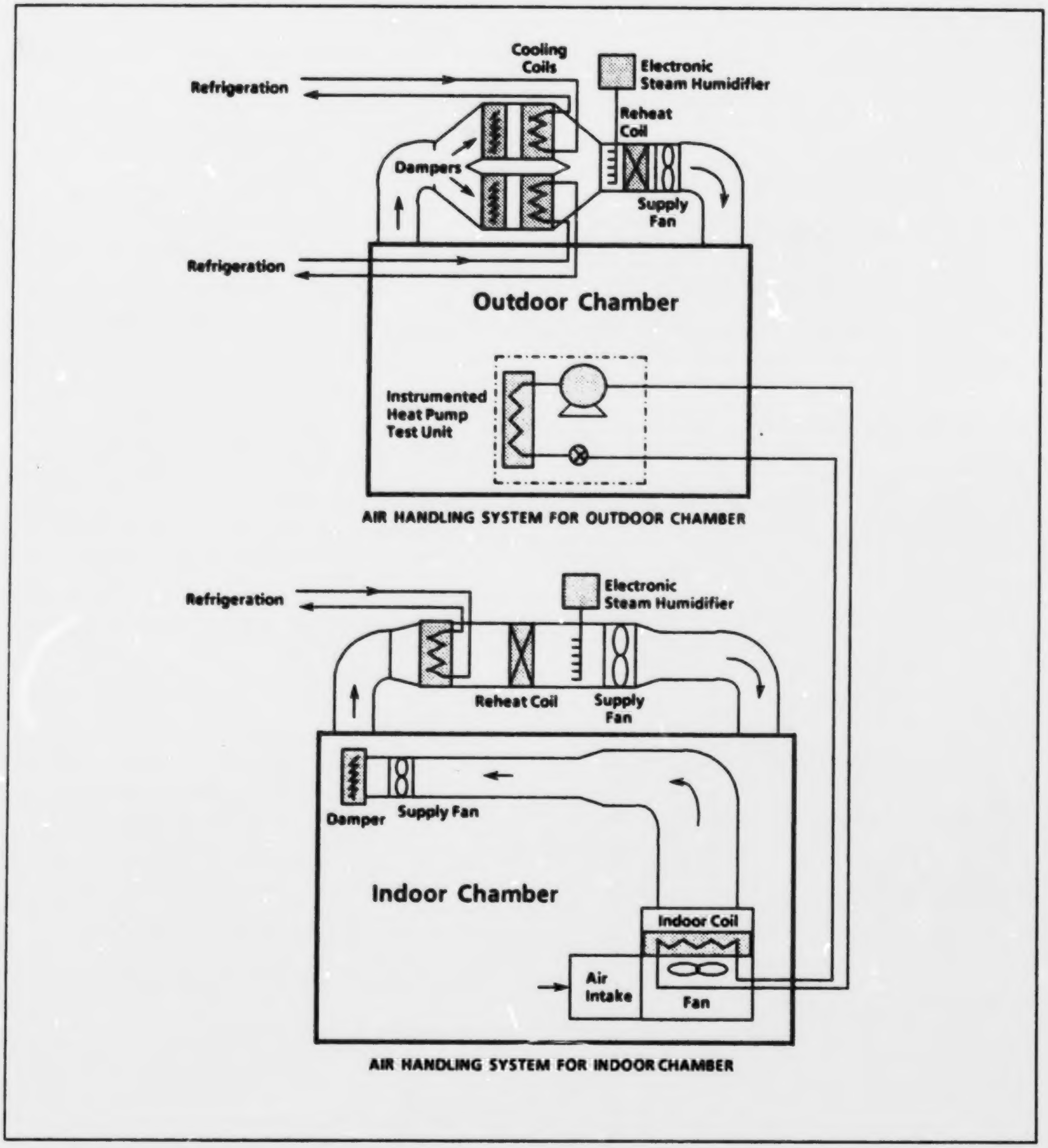

Cooling Coils
Electronic Steam Humidifier
Refrigeration
Reheat Coil
Dampers
Supply Fan
Refrigeration
Outdoor Chamber
Instrumented Heat Pump Test Unit
AIR HANDLING SYSTEM FOR OUTDOOR CHAMBER
Electronic Steam Humidifier
Refrigeration
Reheat Coil
Supply Fan
Damper
Supply Fan
Indoor Chamber
Indoor Coil
Air Intake
Fan
AIR HANDLING SYSTEM FOR INDOOR CHAMBER

Figure 2. National Research Council - Low Temperature Laboratory Simplified Schematic / Layout of Heat Pump Test Facility

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